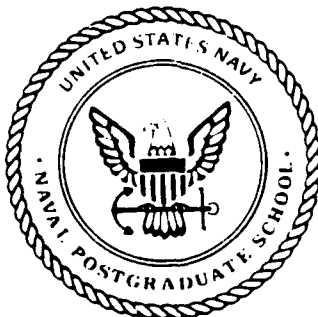


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THESIS

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OPTIMIZATION OF AN INTERNALLY
FINNED ROTATING HEAT PIPE

by

Geraldine L. Olson

June 1992

Thesis Advisor:

D. Salinas

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Optimization of an Internally
Finned Rotating Heat Pipe

by

Geraldine L. Olson
Lieutenant, United States Navy
B.S., United States Naval Academy, 1982

Submitted in partial fulfillment of the
requirements for the degree of

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A.J. Healey, Chairman, Department of
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ABSTRACT

A finite element formulation with linear triangular elements was used to solve the steady-state, two-dimensional conduction heat transfer equation in the condenser wall section of an internally finned rotating heat pipe. A FORTRAN program using this method was coupled with the ADS program for automated design of the internal heat pipe fin geometry to optimize heat transfer. An increase in surface area, which increases heat transfer, also increases the condensate level, which decreases heat transfer. The additional condensate level does not offset the advantage gained by the increased surface area. The investigation provided combinations of fin half angle, number of fins, and fin height for an optimum design. Water is used as the working fluid and the heat pipe is constructed from copper.

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I. INTRODUCTION

A. THE ROTATING HEAT PIPE

The rotating heat pipe is a closed container designed to transfer a large amount of heat in rotating machinery. Since the heat pipe operates on a closed two-phase cycle, the heat transfer capacity is greater than for solid conductors. Also, the thermal response time is less than with solid conductors. The three major elemental parts of the rotating heat pipe are: a cylindrical evaporator, a truncated cone condenser, and a fixed amount of working fluid as shown in figure 1.

An annulus is formed by the working fluid in the evaporator. This occurs at rotary speeds above the critical speed. The addition of heat to the evaporator vaporizes the working fluid. A pressure differential between the evaporator and the condenser causes the vapor to flow towards the condenser. The vapor is transported to the condenser with its latent heat of vaporization. Condensation of the vapor on the inner wall is caused by external cooling. This condensation releases the latent heat of evaporation. This condensate is forced to flow back to the evaporator by a component, acting along the condenser wall, of the centrifugal force which is caused by the rotation of the heat pipe. As the condensate collects in the evaporator the cycle is repeated.

Since the evaporator and condenser portions of a heat pipe function independently, needing only common liquid and vapor streams, the area over which heat is introduced can differ in size and shape from the area

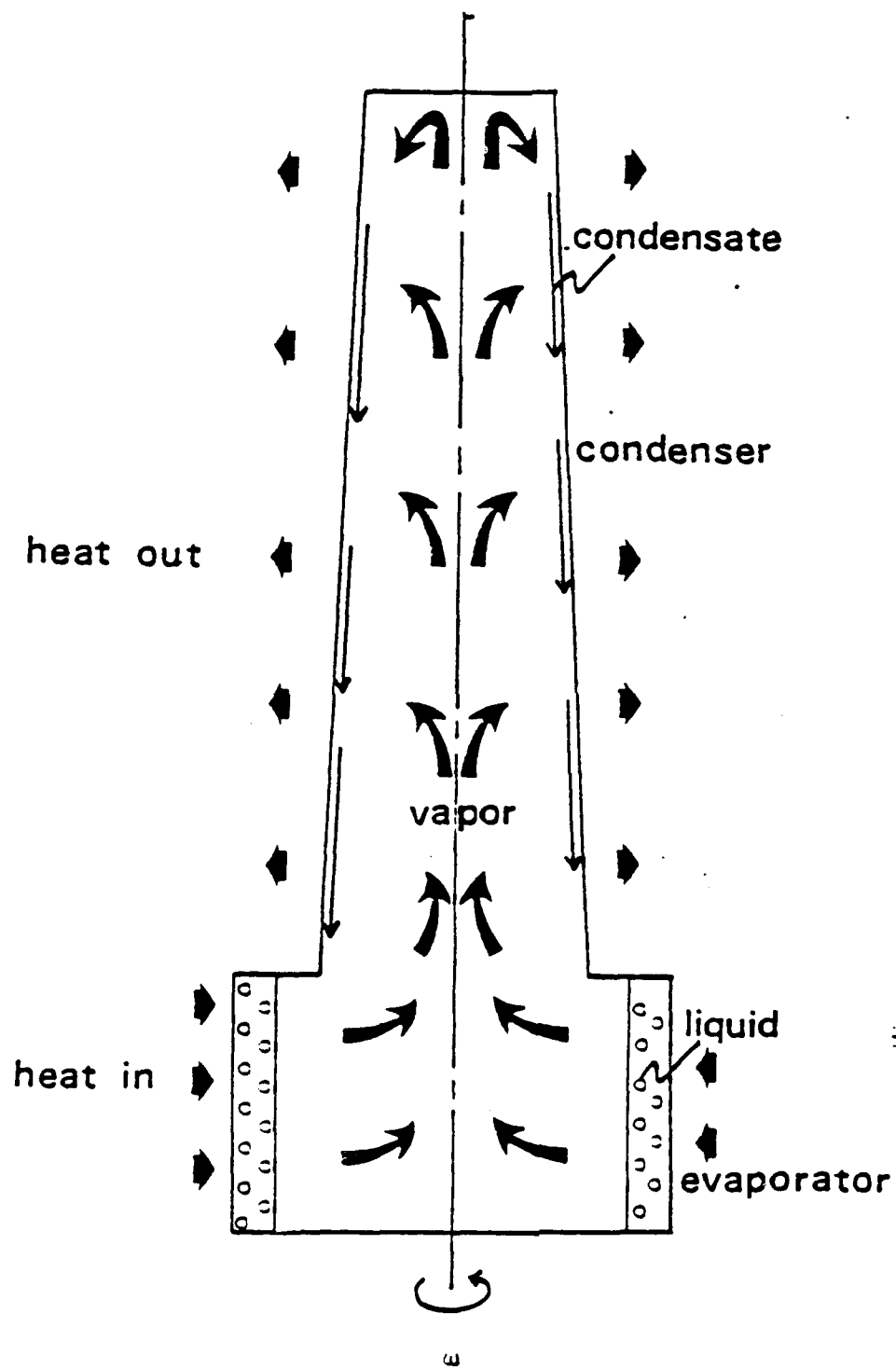


Figure 1. Schematic Drawing of a Rotating Heat Pipe

over which it is rejected, provided that the rate at which the liquid is vaporized does not exceed the rate at which it can be condensed. Therefore, high heat fluxes generated over relatively small areas can be dissipated over larger areas with reduced heat fluxes; allowing a cylindrical evaporator and a truncated cone condenser.

Capillary action acts to drive the condensate back to the evaporator in a conventional heat pipe. No limitation due to capillary action is encountered in a rotating heat pipe nor are external pumps or gravity depended on for the flow of the working fluid. Therefore, the rotating heat pipe can be used in any orientation [Ref. 1].

B. OPERATING LIMITS OF A ROTATING HEAT PIPE

The first theoretical investigation of the rotating heat pipe conducted at the Naval Postgraduate School was performed by Ballback [Ref. 2] in 1969. Various fluid dynamic mechanisms limit the performance of a rotating heat pipe. Ballback [Ref. 2] studied these mechanisms and an estimation of the sonic limit, boiling limit, entrainment limit, and condensing limit of performance was made.

1. The Sonic Limit

The maximum flow of the vapor is set by the choked flow condition in the rotating heat pipe. This limiting vapor flow rate occurs when the heat flux is increased and limits the amount of energy the vapor can transport. The rotating heat pipe effectiveness is reduced due to this limitation. The limiting heat transfer rate due to this condition is:

$$Q_t = \rho_v U_v A h_{fg} \quad (1)$$

and the vapor velocity is considered to be sonic,

$$U_v = c = (g_0 k R T)^{\frac{1}{2}} \quad (2)$$

where

U_v = velocity of the vapor in ft/sec, and

A = cross sectional area for the vapor flow in ft

c = sonic velocity in ft/sec

g_0 = gravitational constant

k = ratio of specific heats

R = gas constant in ft-lbf/lbm R

T = absolute temperature in degrees Rankine

ρ_v = density of vapor in lbm/ft³

2. The Boiling Limit

The transition from nucleate to film boiling was hypothesized by Kutateladze [Ref. 3] to be a completely hydrodynamic process. He determined the following theoretical formula for predicting the burnout flux:

$$Q_t = K \sqrt{\rho_v} A_b h_{fg} \{ \sigma g (\rho_f - \rho_v) \}^{\frac{1}{4}} \quad (3)$$

where

K = constant value

A_b = heat transfer area in the boiler in ft²

h_{fg} = latent heat vaporization in Btu/lbm

σ = surface tension in lbf/ft

g = acceleration of gravity in ft/hr²

ρ_f = density of fluid in lbm/ft³

ρ_v = density of vapor in lbm/ft³.

A constant value for K in the range of 0.13 to 0.19 was suggested by the experimental data obtained by Kutateladze.

3. The Entrainment Limit

The flooding constraint in a wickless heat pipe was examined by Sakhuja [Ref. 4]. The correlation he developed is:

$$Q_t = \frac{A_x C^2 h_{fg} \sqrt{g D (\rho_f - \rho_v) \rho_v}}{\{1 + (\rho_v / \rho_f)^{\frac{1}{4}}\}^2} \quad (4)$$

where

Q_t = heat transfer rate in Btu/hr

A_x = flow rate in ft²

C = dimensionless constant, 0.725 for tube with sharp edged flange

h_{fg} = latent heat of vaporization in Btu/lbm

g = acceleration due to gravity in ft/hr²

D = inside diameter of heat pipe in ft

ρ_f = density of the fluid in lbm/ft³

ρ_v = density of the vapor in lbm/ft³.

4. The Condensing Limit

The condenser section of a rotating heat pipe was modeled as a truncated cone by Ballback [Ref. 2]. Using this model, the condensation limitation for a rotating heat pipe was determined by Ballback [Ref. 2]. He developed the following condensation limit:

$$Q_c = \pi \left\{ \frac{2 K_f \rho_f \omega^2 h_{fg} (T_s - T_w)^3}{3 \mu_f \sin^2 \phi} \right\}^{1/4} [(R_o + L \sin \phi)^{8/3} - R_o^{8/3}]^{3/4} \quad (5)$$

where

Q_c = total heat transfer rate in Btu/hr

k_f = thermal conductivity of the condensate film in Btu/hr-ft-F

ρ_f = density of fluid in lbm/ft³

ω = angular velocity in 1/hr

h_{fg} = latent heat of vaporization in Btu/lbm

T_s = saturation temperature in F

T_w = inside wall temperature in F

μ_f = viscosity of fluid in lbm/ft-hr

ϕ = half cone angle in degrees

R_o = minimum wall radius in ft

L = length along the wall of the condenser in feet.

The geometry and speed of the rotating heat pipe, and the physical properties of the working fluid comprise the condensing limit equation.

For a rotating heat pipe with the physical characteristics as shown in Table I, the amount of heat that can be transferred from the rotating heat pipe is limited by the condensing limit. However, the

limitations imposed by the sonic limit, boiling limit, and entrainment limit may become important as the heat pipe geometry and operating conditions vary.

TABLE I. ROTATING HEAT PIPE SPECIFICATIONS

Length	9.000 inches
Minimum Diameter	1.55 inches
Wall Thickness	0.03125 inches
Internal Half Angle	1.000 degree
Rotating Speed	3600 RPM

To enhance the heat transfer capacity of the rotating heat pipe, internally finned condensers have been used to raise the condensing limit line. Thinner films occur near the ridges of the fins while thicker films occur in the troughs. The thinner film on the ridges provides a lower thermal resistance to heat flow, while the thicker film in the trough provides a higher resistance. A compromise between the improvement on the ridges and the degradation in the troughs is necessary for an overall heat transfer improvement [Ref. 5].

C. ANALYSIS OF THE INTERNALLY FINNED ROTATING HEAT PIPE

Schafer [Ref. 6] developed an analytical model for a heat pipe with a triangular fin profile (figure 2). This model was developed in order to raise the condensing limit by the addition of internal fins. An assumption of one-dimensional heat conduction through the wall and fin was made for Schafer's model.

A two-dimensional heat conduction model using a Finite Element Method was developed for this same case by Corley [Ref. 7]. A parabolic

temperature distribution along the fin surface was assumed by Corley [Ref. 7]. A significant improvement in the predicted heat transfer performance was indicated by his results. By using the two-dimensional model an increase of approximately 75 percent in the heat transfer performance was seen over the results from the use of the one dimensional model. However, Corley [Ref. 7] noted that at the fin apex an error as great as 50 percent was possible which could result in the total heat transfer being in error as much as 15 percent.

A modification was made to Corley's computer program by Tantrakul [Ref. 8]. In order to minimize the heat transfer error at the apex of the fin Tantrakul increased the number of finite elements used. His results with this modification converged with the results of Corley.

Purnomo developed a linear triangular finite element model (figure 3) used in a two-dimensional Finite Element Method solution. Purnomo's [Ref. 1] Finite Element Method program also worked and converged. To maximize the heat transfer from the rotating heat pipe the condenser geometry was varied. Using Purnomo's code parametric studies were conducted. However, the best geometry was not indicated in these studies. Purnomo's code was written to perform one analysis at a time. Davis [Ref. 9] modified Purnomo's code to allow for numerous analysis to be made using the optimization code COPES/CONMIN. Davis' Finite Element Method code incorporating the optimization worked and converged, resulting in an optimum design for an internally finned rotating heat pipe.

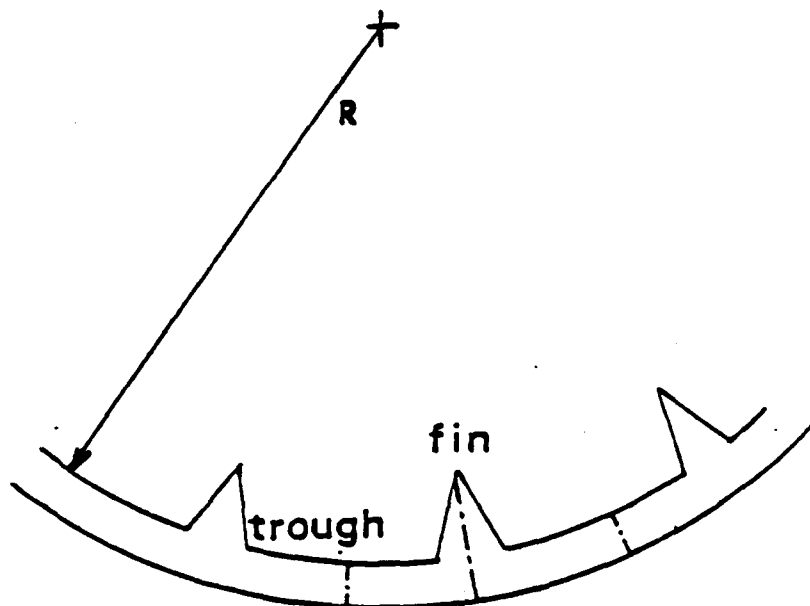


Figure 2. Internally Finned Condenser Geometry, Showing Fins, Troughs and Lines of Symmetry.

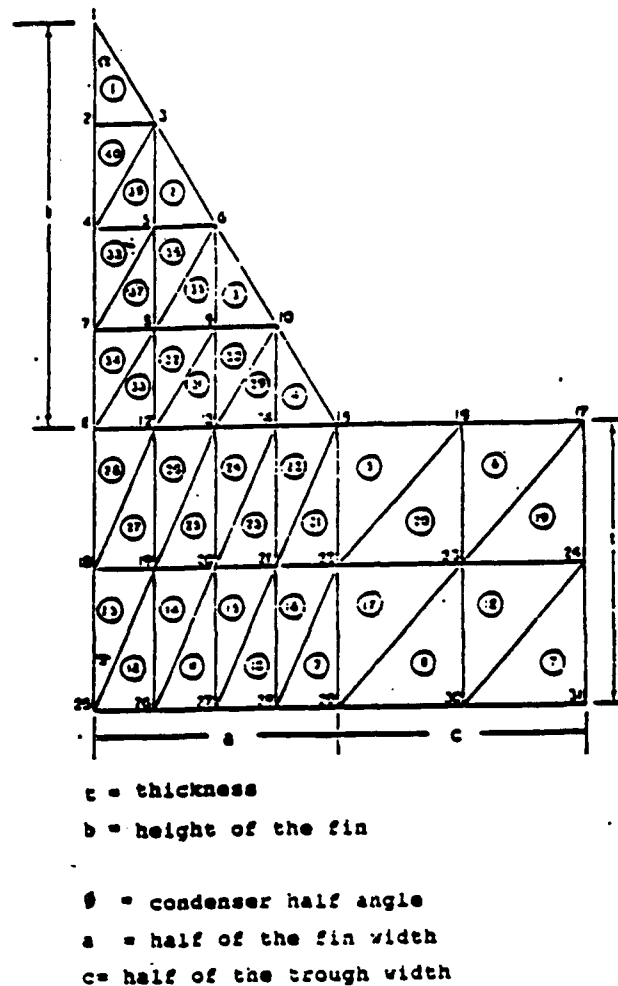


Figure 3. Condenser Geometry Considered with 40 Linear Triangular Finite Elements.

D. THESIS OBJECTIVES

The objectives of this thesis were:

1. To modify Davis' [Ref. 9] computer program so that it is compatible with the ADS (Automated Design Synthesis) program [Ref. 10] and can be used for analysis and automated design of rotating heat pipes.

2. To use the resulting program to obtain an optimum design for an internally finned rotating heat pipe to obtain experimental data to compare with the analytical results.
3. To use the resulting program to obtain numerical results in place of data obtained from costly experimental operations.

II. NUMERICAL OPTIMIZATION

A. BACKGROUND

The parameter that is minimized or maximized during the design process is called the design objective. The design objective is minimized or maximized by changing the design variables within the design constraint limitations. This process is called numerical optimization. An assortment of physical, aesthetic, economic and, on occasion, political limitations must be met by the design constraints for the design to be acceptable. For the optimization process to work, the design criteria must be described in numerical terms. This is not always easy.

A computer program can be written to perform tedious and repetitive calculations necessary to optimize the problem once it is stated in numerical terms. For this reason, computer analysis is commonplace in most engineering organizations. For example, in heat transfer design the configuration, materials, and method of heat removal may be defined and a finite element analysis computer code is used to calculate temperatures, heat transfer rates, and other response quantities of interest. If any of these parameters are not within prescribed bounds, the engineer may change the method of cooling or other defined quantity and rerun the program. The engineer makes the actual design decisions, the computer code only provides the analysis of a proposed design. This is the commonly used approach which is called computer-aided design.

Analysis codes are commonly used for tradeoff studies. For example, an analysis code might be run on the distance a truck can go on a tank of fuel. For different loads, different distances are calculated which can be used in a range-payload study.

Fully automated design is the logical next step to computer-aided design. The computer makes the actual design decisions or trade-off studies based on input criteria in fully automated design. Minimal information is requested from the operator during the actual design process. Numerical optimization offers numerous improvements over the traditional approach to design. These improvements include: time reduction in design decision making; a rational, directed design procedure; and the procedure is unbiased by intuition or experience. The probability of obtaining a non-traditional solution is thereby improved. Engineering intuition and experience are still necessary to decide if the design obtained is an improvement and feasible.

B. AUTOMATED DESIGN SYNTHESIS (ADS)

Vanderplaats [Ref. 10] developed a general purpose numerical optimization program containing a variety of algorithms, ADS. ADS is a FORTRAN program that optimizes a numerically defined objective function subject to a set of constraint limits. The solution of the problem is separated into three levels:

1. Strategy - Optimization strategy such as Augmented Lagrange Multiplier method or Sequential Linear Programming.
2. Optimizer - Actual algorithm to perform the optimization.
3. One-Dimensional Search - Line search routine used by optimizer.

Flexibility to solve a wide variety of engineering design problems is given by the combinations of nine strategies, five optimizers, and eight one-dimensional search options. The following definitions are necessary to discuss the use of ADS:

1. **Design Variables** - Those parameters which the optimization program is permitted to change within allowed bounds in order to improve the design. Design variables appear only on the right hand side of an equation and are continuous.
2. **Design Constraints** - An inequality constraint requires that some function of the design variable(s) remain less than a specified value. Design constraints may be linear or nonlinear, implicit or explicit, but they must be continuous functions of the design variable.
3. **Objective Function** - The parameter which is going to be minimized or maximized during the optimization process. The objective function may be linear or nonlinear, implicit or explicit, and must be a continuous function of the design variables. The objective function usually appear on the left side of an equation.

C. PROGRAMMING GUIDELINES

Any computer code developed for engineering analysis should be written in such a way that it is easily coupled to a general purpose optimization program such as ADS. Therefore, a general programming practice is outlined here which in no way inhibits the use of the computer program in its traditional role as an analytical tool, but allows for simple adaption to ADS.

ADS is called by a user-supplied calling program. ADS does not call any user-supplied subroutines. Instead, ADS returns control to the calling program when function or gradient information is needed. The required information is evaluated and ADS is called again. This provides considerable flexibility in program organization and restart capabilities. Various internal

parameters are defined on the first call to ADS which work well for the "average" optimization task. However, it is often desirable to change these in order to gain maximum utility of the program. Figure 4 is the program flow diagram for the case where the user wishes to over-ride one or more internal parameters, such as scaling, convergence criteria, or maximum number of iterations.

After initialization of basic parameters and arrays, the information parameter, INFO, is set to -2. ADS is then called to initialize all internal parameters and allocate storage space for internal arrays. Control is then returned to the user, at which point these parameters, for example convergence criteria, can be overridden if desired. At this point, INFO will have a value of 1 and the user must evaluate the objective function, OBJ, and constraint functions. ADS is called again and the optimization proceeds. Since, in this case, the gradient calculation control, IGRAD, has a value of zero, all gradient information is calculated by finite difference within ADS. When INFO has a value of zero, optimization is complete.


```

      BEGIN
      DIMENSION ARRAYS
      DEFINE BASIC VARIABLES
      IGRAD=0 (USE FINITE DIFFERENCE GRADIENTS)
      INFO = -2
      CALL ADS (INFO...)
      IF INFO = 0, EXIT. ERROR WAS DETECTED
      ELSE
      OVER-RIDE DEFAULT PARAMETERS IN ARRAYS WK AND IWK IF
      DESIRED
      CALL ADS (INFO...)

      NO                                     YES

      INFO = 0

      EVALUATE OBJECTIVE                     EXIT OPTIMIZATION
      AND CONSTRAINTS                       IS COMPLETE

```

Figure 4. Program Flow Logic: Over-Ride Default Parameters, Finite Difference Gradients [Ref. 10].

III. FINITE ELEMENT SOLUTION

A. REVIEW OF THE PREVIOUS ANALYSIS

As stated previously, the heat transfer solution for a one-dimensional model of an internally finned rotating heat pipe was studied by Schafer [Ref. 6]. The two-dimensional model was studied by Corley [Ref. 7]. The same assumptions and boundary conditions, similar to those used in the Nusslet analysis of film condensation on a flat wall, and based upon the analysis of Ballback [Ref. 2] were used for both. The more important of these assumptions are:

1. steady state operation,
2. film condensation, as opposed to dropwise condensation,
3. laminar flow of the condensate film along both the fin and the trough,
4. static balance of forces within the condensate,
5. one-dimensional conduction heat transfer through the film thickness (no convective heat transfer in the condensate film),
6. no liquid-vapor interfacial shear forces,
7. no condensate subcooling,
8. zero heat flux boundary conditions on both sides of the wall section (symmetry conditions), as shown in figure 5,
9. saturation temperature at the fin apex,
10. zero film thickness at the fin apex, and
11. negligible curvature of the condenser wall.

Figure 3 shows the linear triangular finite element model developed by Purnomo [Ref. 1] for use in obtaining a two-dimensional Finite Element solution.

The assumption that was used by Corley [Ref. 7] that the fin apex was at the saturation temperature of the working fluid was modified by Purnomo [Ref. 1]. The value of the temperature at the apex of the fin was allowed to float and a parabolic temperature distribution was assumed along the fin surface.

Purnomo's problem statement for the formulation of the Finite Element Method as shown in figure 5 is:

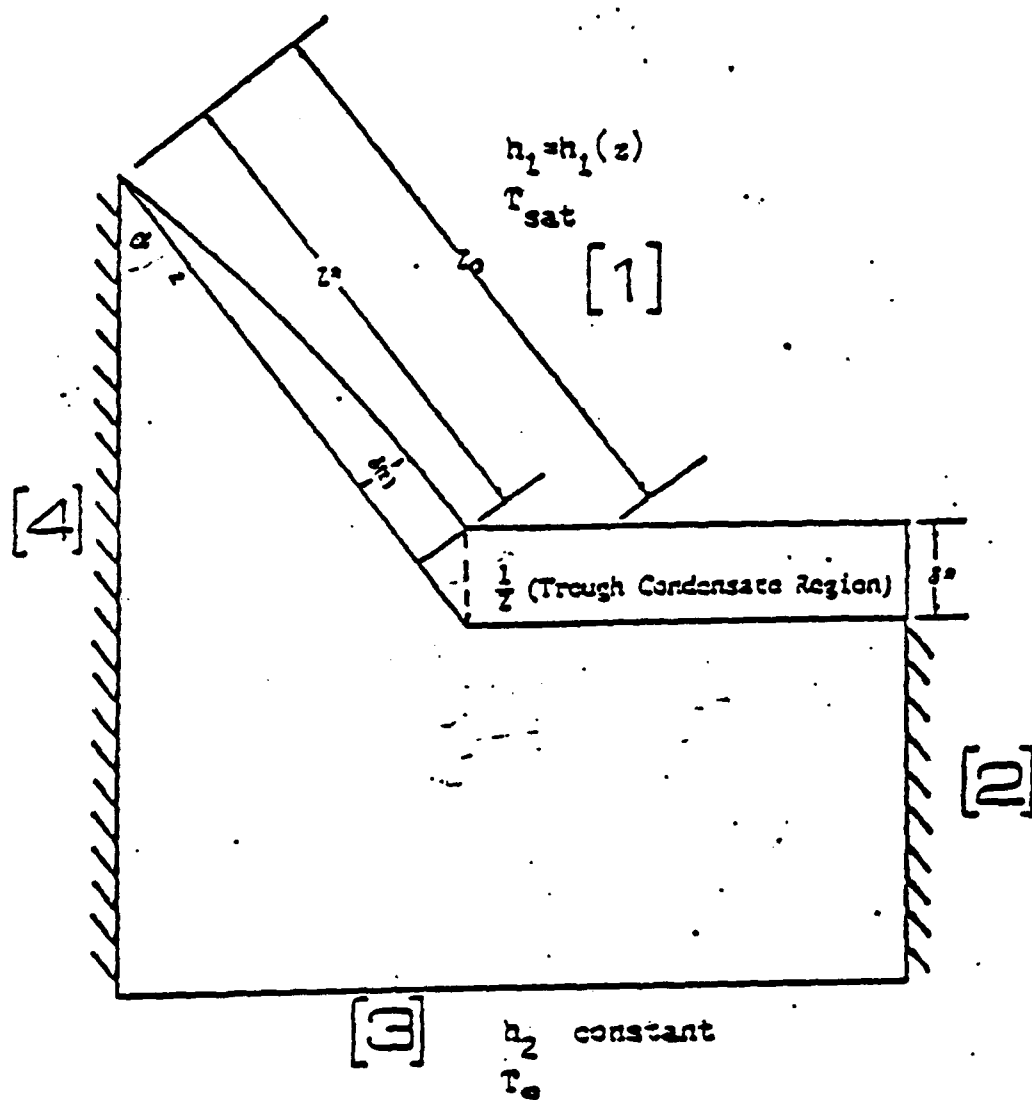
$$\frac{\partial^2 T}{\partial X^2} + \frac{\partial^2 T}{\partial y^2} = 0 \quad (6)$$

with the following boundary conditions:

1. along boundary 1, $-K \partial T / \partial n = h_1 (T - T_{sat})$
2. along boundary 3, $-K \partial T / \partial n = h_2 (T - T_{\infty})$
3. along boundaries 2 and 4, $\frac{\partial T}{\partial n} = 0$

A detailed description of the numerical formulation is presented in his thesis.

Davis used Constrained Function Minimization (CONMIN) as an optimization program. CONMIN is a FORTRAN program in subroutine form.



$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = 0$$

b.c.

- a) $-k \frac{\partial T}{\partial n} = h_1 (T - T_{sat})$ Along Boundary [1]
- b) $-k \frac{\partial T}{\partial n} = h_2 (T - T_\infty)$ Along Boundary [3]
- c) $\frac{\partial T}{\partial n} = 0$ Along Boundaries [2] and [4]

Figure 5. Differential Equation and Boundary Conditions Considered in the Analysis of Purnomo [Ref. 1].

Vanderplaats [Ref. 11] developed the Control Program for Engineering Synthesis (COPES) as a main program to simplify the use of CONMIN. Davis' computer program was written in subroutine form with SUBROUTINE ANALIZ (ICALC) as the main routine. The name ANALIZ is compatible with the COPES program and ICALC is a calculation control. Subroutine ANALIZ calls other subroutines as needed:

1. the routine "COORD" used to define positions of system coordinate points,
2. the routine "FORMAF" used to formulate the Finite Element Method equations,
3. the routine "BANDEC" as an equation solver for a symmetric matrix which has been transformed into banded form, and
4. the routine "DPLORT" used to compute the roots of a real polynomial using a Newton-Raphson derivative technique.

B. THE COMPUTER PROGRAM DEVELOPMENT

The basis for the present analysis code is Davis' [Ref. 9] two-dimensional finite element program. Davis' code was checked for validity as the first task undertaken in the development of this thesis. An error was discovered in calculating the fin condensate film thickness (AZS). In the initial calculation of HDEN only, the cubed term was merely multiplied by three. The correct form of the equation is shown below:

$$HDEN = -a_1 z^3/3 - b_1 z^2/2 + z(T_{sat} - T_1)$$

The effect of this error was minimal since subsequent equations were correct, a 0.00016% difference in the condensate level was noted.

The next task undertaken was to adapt the analysis code to permit automated design analysis using ADS. Many modifications were made, some of which are mentioned here. The program was rewritten to include a main program from which ADS is called and subroutines to perform various mathematical functions. Subroutine ANALIZ was deleted since the double precision version of ADS (DADS) was used. Modifications were also made to generalize the code and minimize the changes needed when varying the number of finite elements used.

A listing of the revised computer program is included as the Appendix.

C. DESIGN OPTIMIZATION

There are thirteen parameters that can be used as design variables. There are geometric or functional parameters of the rotating heat pipe or the properties of the working fluid or environment. The possible design variables, possible constraint functions, and the objective function are listed below in Fortran. This code can pursue a wide variety of design problems.

The addition of fins by the designer increases the surface area which increases the heat transfer rate through the condenser wall. However, the addition of fins decreases the cross-sectional area through each fin for conduction and decreases the trough width which increases the condensate thickness in the trough. The increased condensate thickness decreases the heat transfer and if increased to the point of covering the fins it could dramatically reduce the heat transfer through the fin.

TABLE II. DESIGN VARIABLES

DESIGN VARIABLES
BFIN (fin height) CANGL (cone half angle) CLI (condenser length) FANGL (fin half angle) HINF (external convective heat transfer coefficient) NFIN (number of fins) R2I (intermediate radius) RBASEI (inside radius of condenser base) RMP (rotational speed of the heat pipe) THICKI (condenser wall thickness) TINF (external temperature) TSS (saturation temperature) TZ (nodal point temperature)
CONSTRAINT FUNCTIONS
BOA (ratio of fin height over fin width) ZOA (ratio of trough width over fin width) DEL(NI) (condensate thickness)
OBJECTIVE FUNCTION
OBJ

The purpose of the design study was to determine the fin height, number of fins, and fin half angle which would maximize the heat transfer rate. It was then decided that the design variables would be BFIN, FANGL, and NFIN. The number of fins was chosen vice the ratio of trough width to fin width (ZOA) as was previously used. This decision was made since it is easier to think in terms of the number of fins vice a ratio. Other potential design variables were held constant. The objective function to be maximized was $OBJ = QT + QTF$, the heat transfer through the fin plus the heat transfer through the trough.

The code was run with the three design variables using the internal scalar default parameters in ADS. The objective function was calculated using the input values of the design variables. The ADS program then

changed the first design variable keeping the other two design variables at the input values. The calculations were made aging yielding a different value for the objective function. The new value would then be compared to the previous value of the objective function. If the difference in the objective function was not greater then the internal scalar default value, the first design variable would be returned to its original value and the process repeated with the second design variable. If the difference in the objective function was still not greater then the internal scalar default value, the second design variable would be returned to its initial value and the process repeated with the third design variable.

Each time the program was run, the optimization code would choose BFIN as the design variable to change first as it had the greater influence on the objective function. The remaining two variables would then either be kept constant or the number of fins would be maximized and the fin half angle minimized. Consistent results were not obtained with this method.

To improve the results, the internal scalar parameters were modified. These modifications included the constraint tolerances, the absolute and relative convergence criteria, the absolute and relative change in the design variables, the absolute and relative change in the objective function, the minimum absolute value of the finite difference step when calculation gradients, and the initial relative move limit. Better results were obtained as seen in the higher value for the objective function. However, the results were still not consistent depending on the initial values used for the three design variables. At this point, it was decided to concentrate on one design variable and on the basis of the previous calculations the design variable chosen was BFIN.

The external surface temperature was set equal to the working fluid saturation temperature and the theoretical upper limit on the heat transfer was calculated for comparison. An assumption is made that there is no thermal resistance across the condensate or the condenser wall. The upper limit of the heat transfer rate was predicted to be 69,492 BTU/HR using the following formula:

$$Q_{\max} = 2\pi\bar{r}l(T_{\text{wall}} - T_a) \quad (7)$$

where

h = outside convective heat transfer coefficient (5000 BTU/HR·FT²·F)

\bar{r} = average outside radius of condenser wall (0.07373 FT)

l = condenser length (0.75 FT)

T_{wall} = temperature of the outside wall (100°F)

T_a = ambient temperature (60°F)

Based on engineering judgement certain constraints were placed on the design. These constraints were applied to the number of fins (not to exceed 400) and the minimum fin half angle (not to be less than 10 degrees). The values were based on structural and manufacturing considerations.

IV. RESULTS

A. INTRODUCTION

The purpose of the design optimization was to maximize the heat transfer rate. This was accomplished by using the computer code in conjunction with ADS. Numerical results are discussed below.

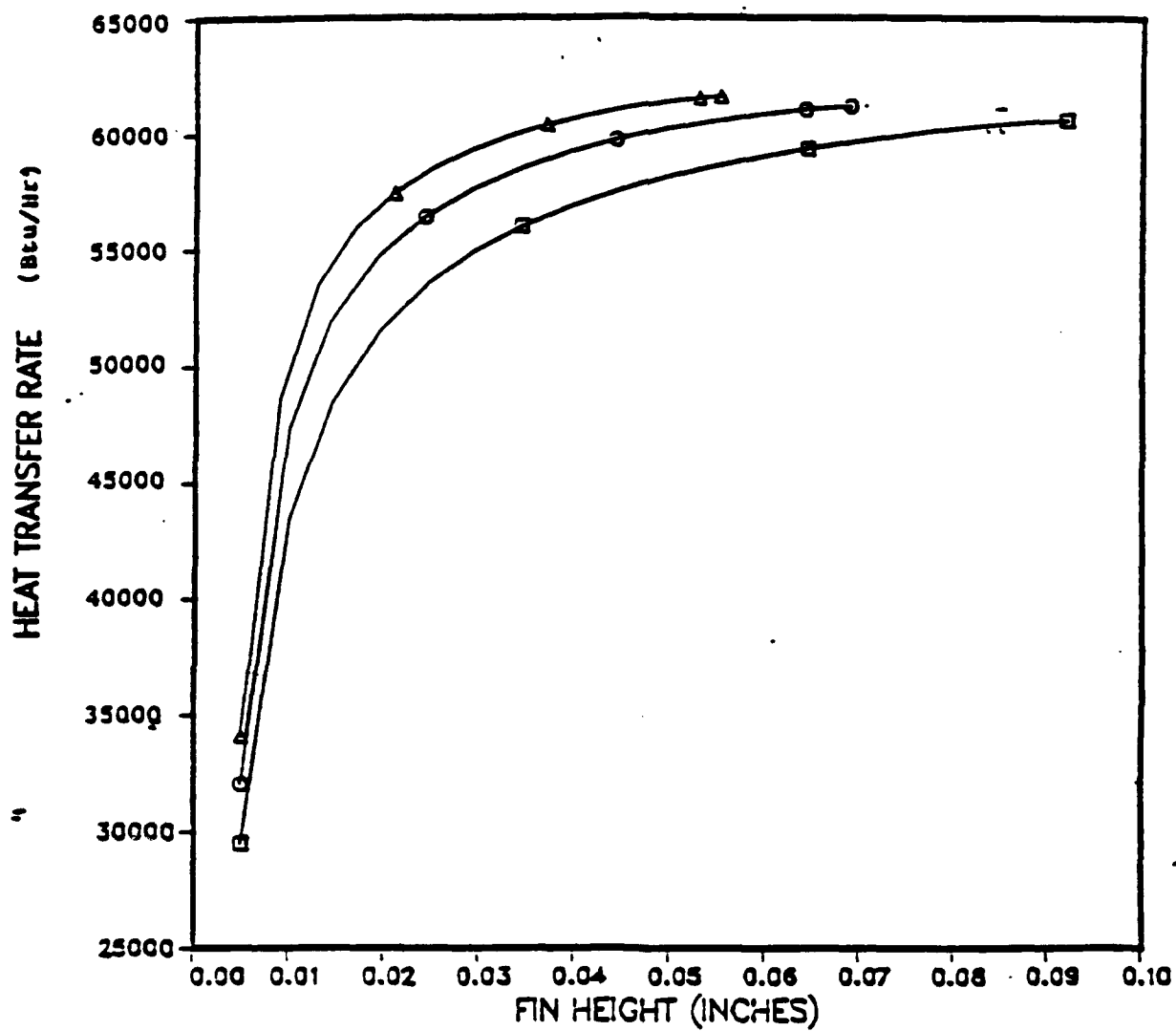
B. CONSTRAINED OPTIMIZATION

In the design problem undertaken to determine the optimum internal geometry for the maximum heat transfer, numerous runs were made for a condenser made of copper. This material has a thermal conductivity of 230 BTU/HR'FT'F. The working fluid was water.

Since the fin height (BFIN) was the design variable, the initial runs investigated whether there was an optimum fin height. Initially the fin half angle was held constant at 10 degrees and the number of fins was varied from 150 to 400. In each case, for the optimum design, the fin height was maximized and the trough was eliminated, as seen in figures 6 and 7.

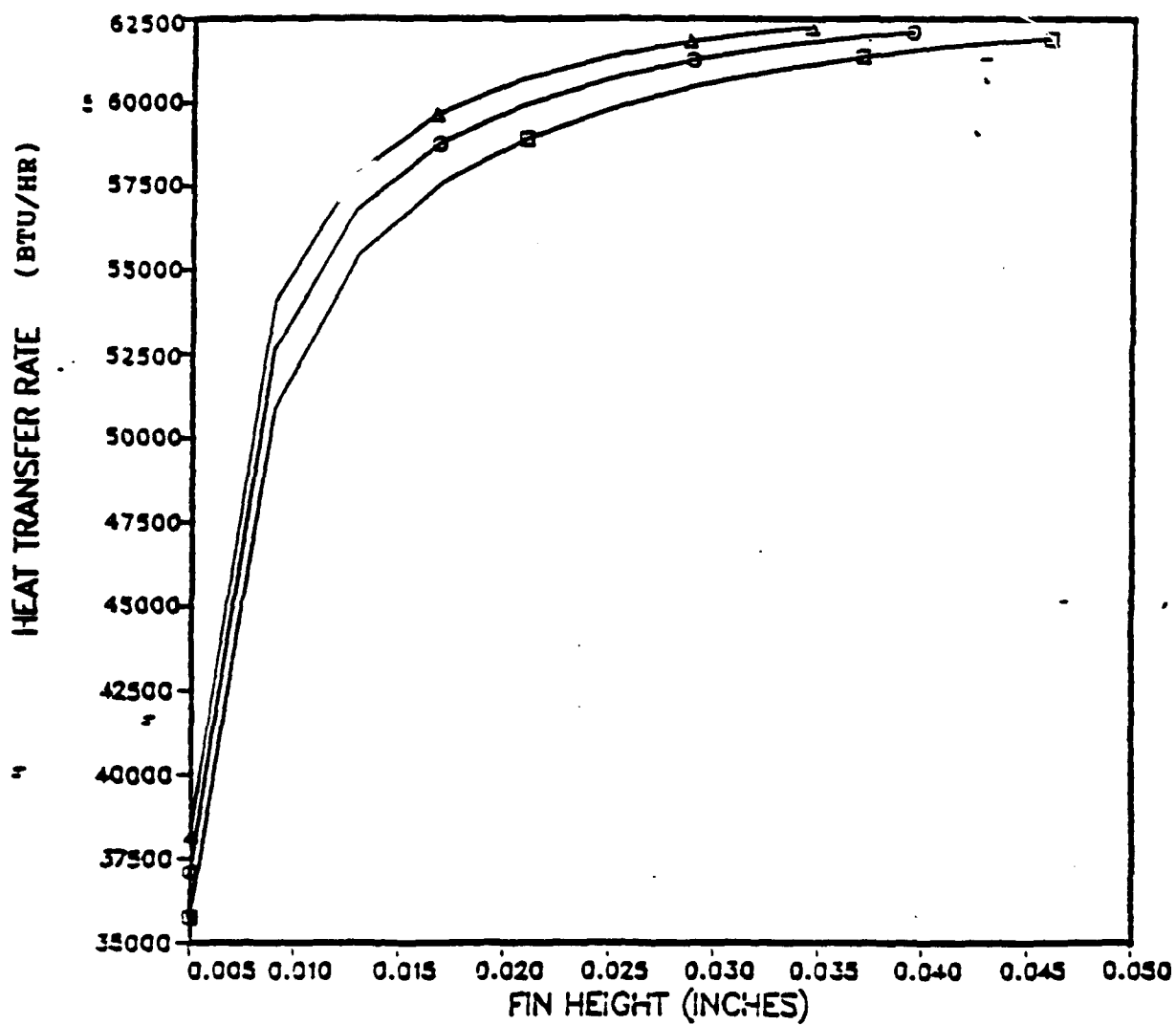
The number of fins was then held constant and the fin half angle was varied from 10 to 25 degrees with the fin height remaining the design variable. Once again, the greatest heat transfer rate was achieved with the highest fin height for each number of fins (figure 8.,

As seen in figure 9, the highest heat transfer rate achieved was for 400 fins with a 10 degree fin half angle and a fin height of 0.0345 inches.



LEGEND
□ = NUMBER OF FINS=150
○ = NUMBER OF FINS=200
△ = NUMBER OF FINS=250

Figure 6. Heat Transfer Rate vs. Fin Height.



LEGEND
□ = NUMBER OF FINS=300
○ = NUMBER OF FINS=350
△ = NUMBER OF FINS=400

Figure 7. Heat Transfer Rate vs. Fin Height.

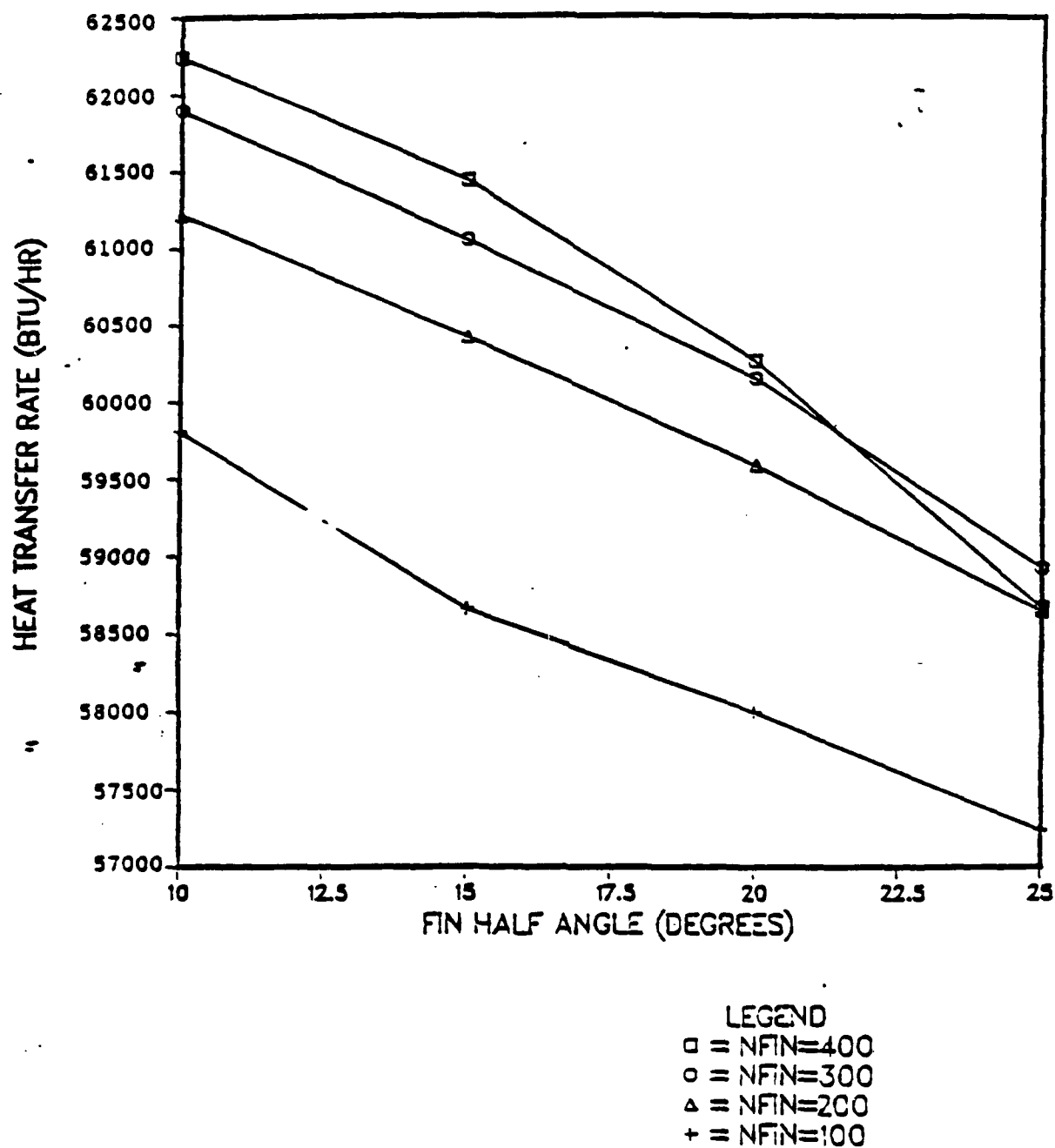


Figure 8. Heat Transfer Rate vs. Fin Half Angle (Optimum Fin Height).

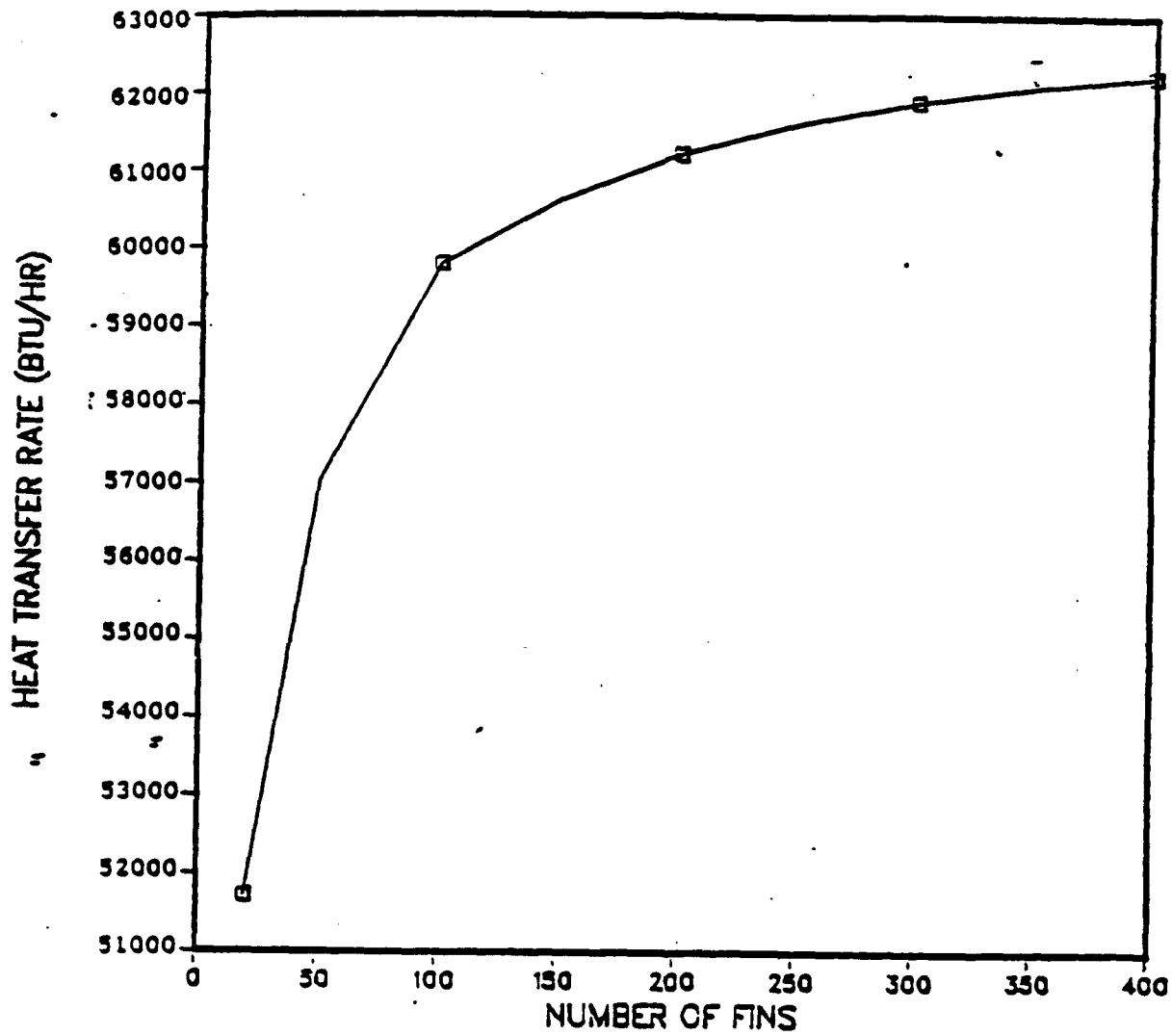


Figure 9. Heat Transfer Rate vs. Number of Fins (Optimum Design).

Figure 10 shows a plot of heat transfer rate versus fin half angle for a condenser with between 100 and 400 fins. The heat transfer rate, as a function of fin half angle for a constant fin height, increases as the fin half angle increases. Davis [Ref. 9] concluded that the heat transfer rate increased with an increase in fin half angle. This is correct if the fin height is kept constant. As the fin half angle increases, the surface area of the fin increases. The added surface area also has a thinner film of condensate on it which offers lower thermal resistance. The trough area decreases and the condensate film in the trough thickens, increasing the thermal resistance. This degradation does not offset the gain in the heat transfer rate caused by the fin.

For external heat transfer coefficients varying from 1000-50,000 BTU/HR·FT²·F, the same optimum design geometry for a maximum heat transfer rate was obtained, which is stated in table III below. Figure 11 shows the strong influence the external heat transfer coefficient has on the heat transfer rate.

In figure 12, the effect of the rotating speed on the heat transfer rate is seen. As the rotational speed increases, the heat transfer rate increases, this is caused by an increase in the element heat transfer coefficient and a decrease in the condensate thickness on the fin which lowers the thermal resistance.

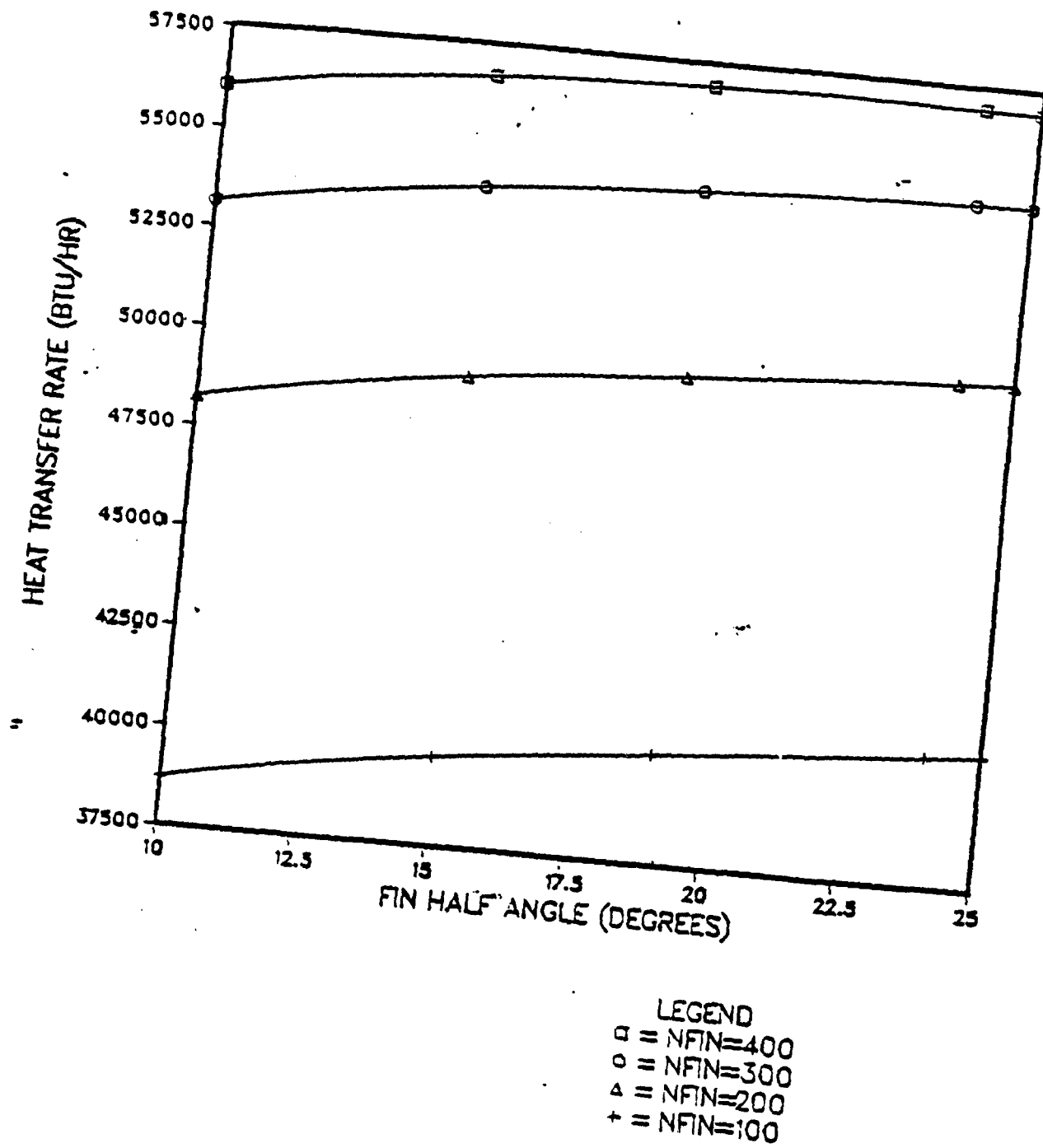


Figure 10. Heat Transfer Rate vs. Fin Half Angle (Constant Fin Height).

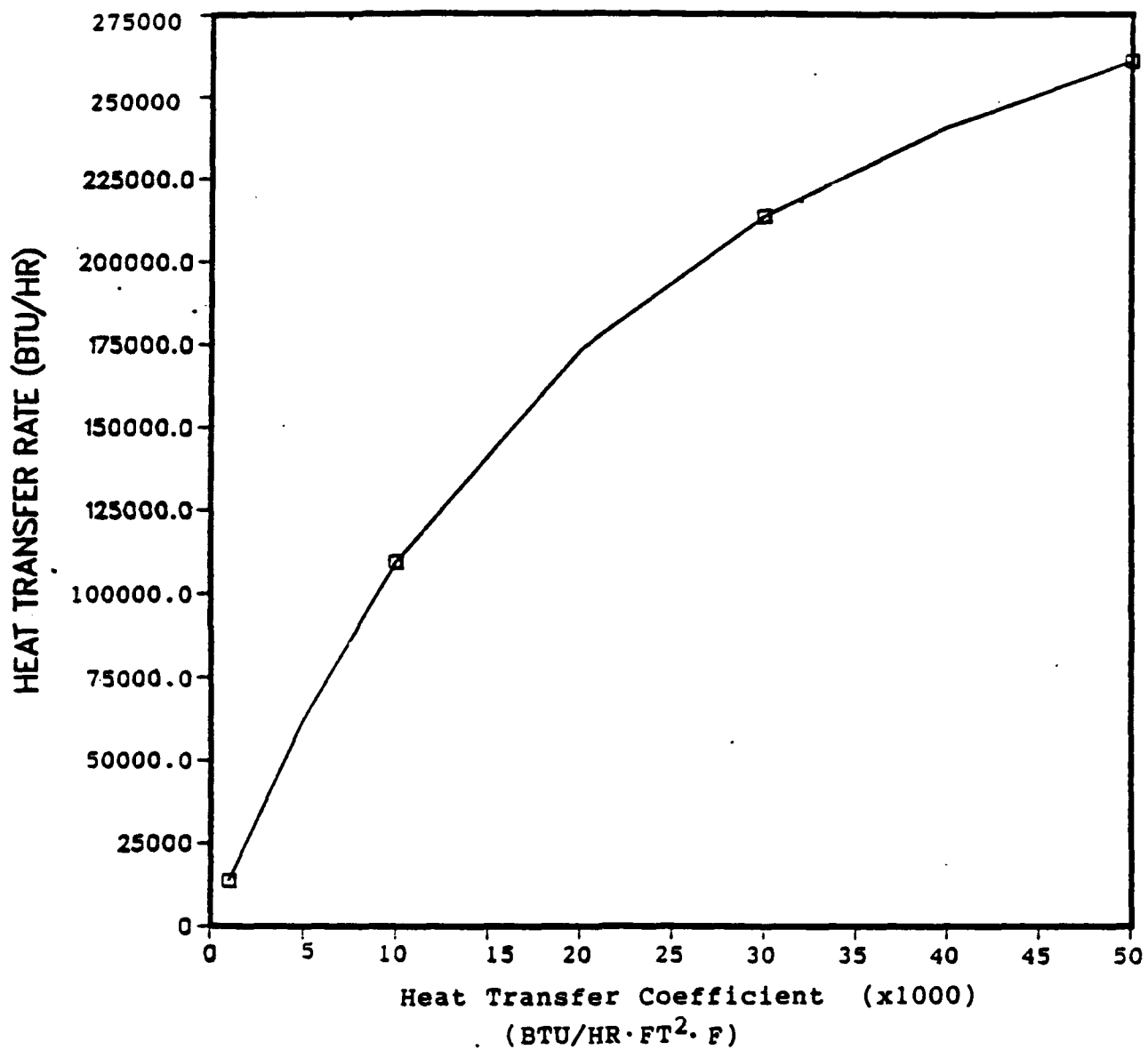
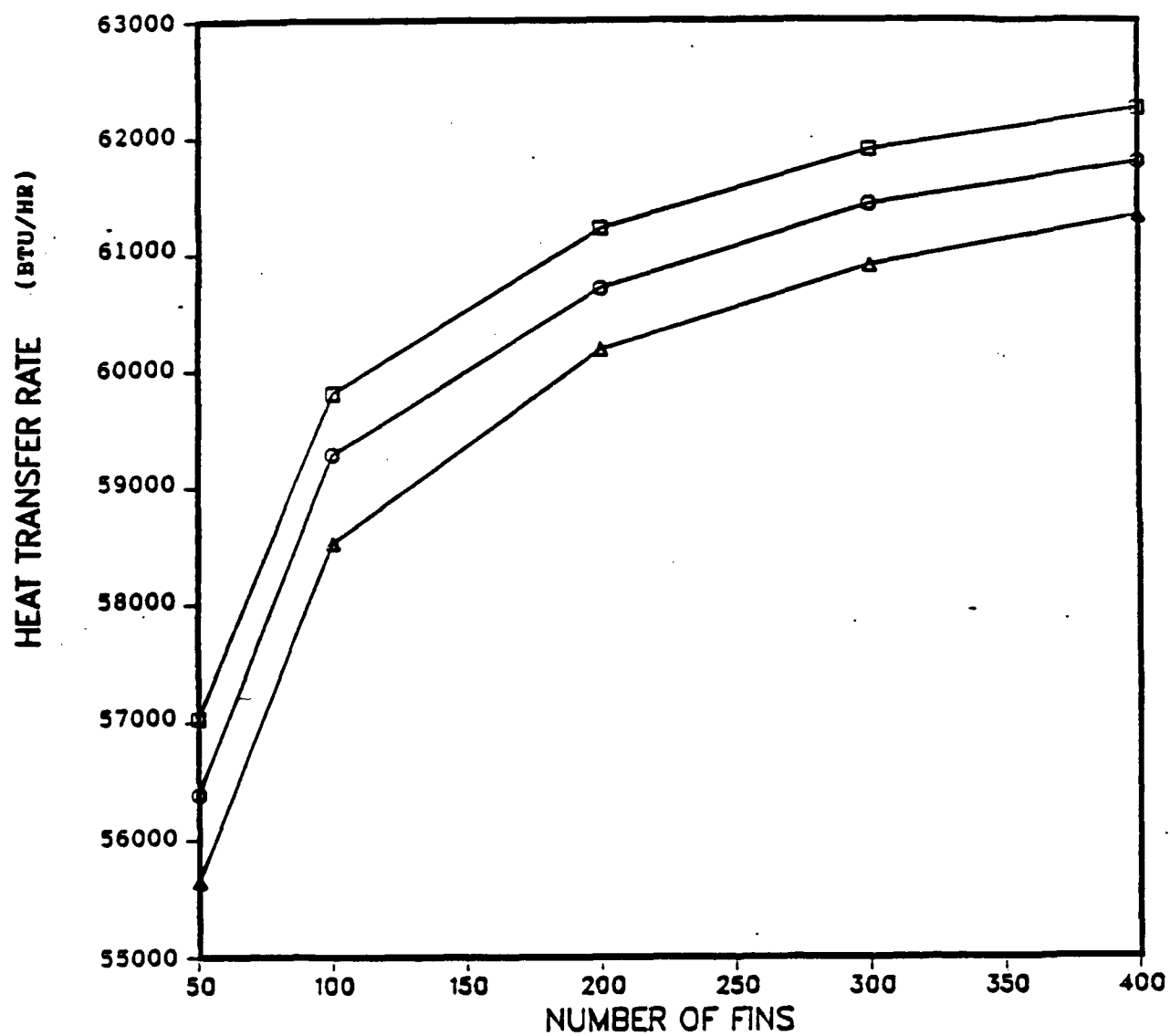


Figure 11. Heat Transfer Rate vs. Heat Transfer Coefficient.



LEGEND
□ = RPM= 3600
○ = RPM= 3000
△ = RPM= 2500

Figure 12. Heat Transfer Rate vs. Number of Fins (RPM Variation).

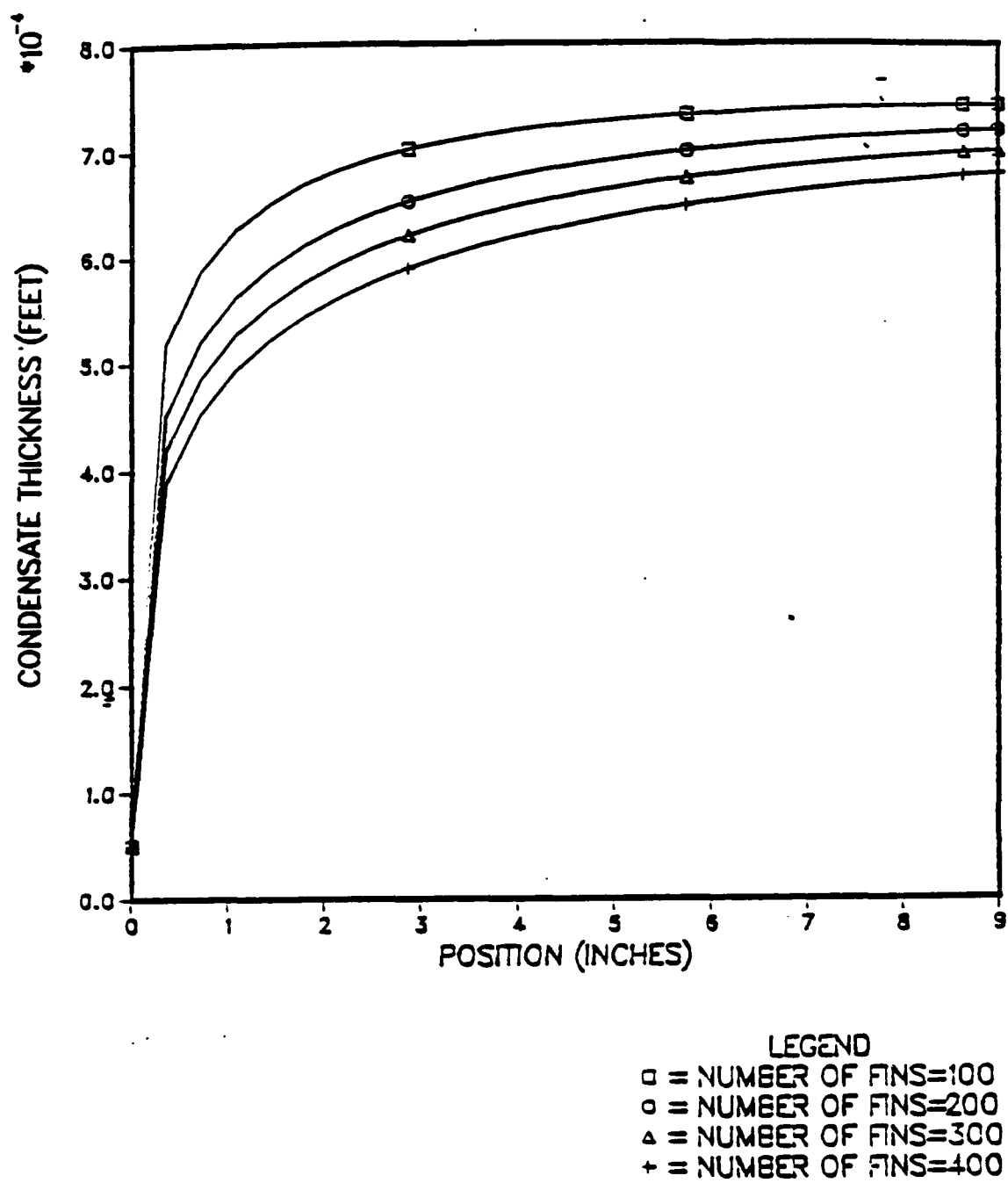


Figure 13. Condensate Level vs. Position (100-400 Fins).

When the number of fins was increased from 100 to 400, maintaining the same fin half angle, the condensate level decreased with the increased heat transfer rate (figure 13). This occurred because the thinner film over the fins decreased the resistance across the film which raised the temperatures along the fin, which in combination with the lower height fins increased the temperature on the outside of the pipe. This increase in temperature brings the operation closer to the condensing limit. When the fin half angle was increased for a specified number and height of fins, the condensate level decreased due to the increased trough width (figure 14).

**TABLE III. OPTIMIZATION RESULTS FOR
VARIOUS HEAT TRANSFER COEFFICIENTS**

OPTIMUM DESIGN	
Fin Height Fin Half Angle Number of Fins	0.0345 inches 10.0 degrees 400
h_{external} (BTU/HR·FT ² ·F)	HEAT TRANSFER RATE (BTU/HR)
1000	13765
5000	62252
50000	261003

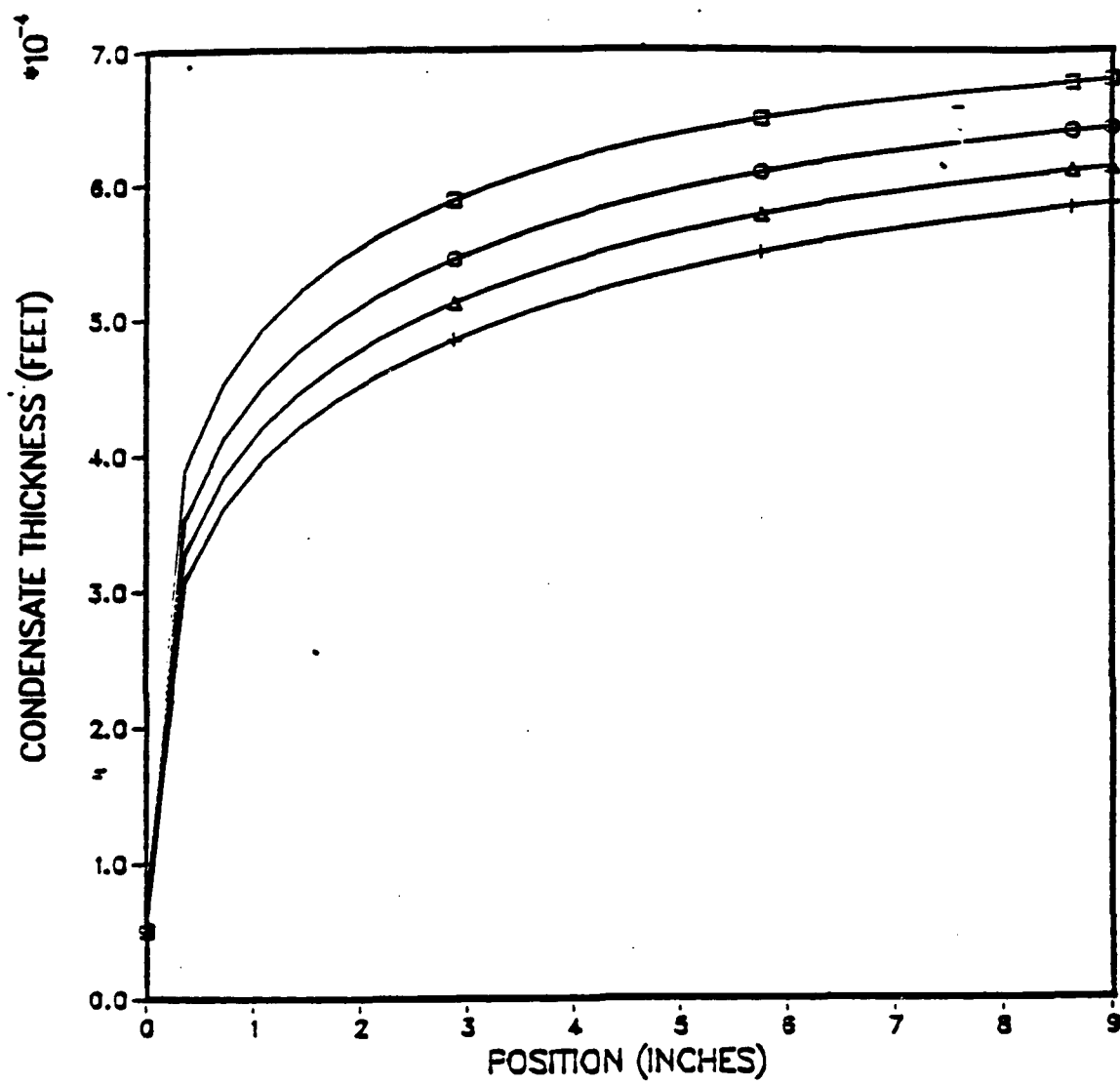
Figures 15 and 16 show the effect the increase of surface area has on the heat transfer rate. In figure 15, the fin height for 400 fins with a 10 degree half angle is plotted against the heat transfer rate. As the fin height is increased up to a maximum value of 0.0345 inches the resulting design is a sawtooth. Figure 16 shows the effect adding more fins has on the heat transfer rate for a constant height fin. The greatest increase in

the heat transfer rate is seen from the addition of 100 fins from a smooth tube.

In figure 17, the ratio of the actual surface area over the surface area for a smooth tube is plotted versus the number of fins. The fin height and the fin half angle are both held constant. As expected, the surface area ratio increases in a linear manner as the number of fins increases. The ratio of the heat transfer rate over the heat transfer rate for a smooth tube is plotted for two different heat transfer coefficients, $h=1000 \text{ BTU/HR}\cdot\text{FT}^2\cdot\text{F}$ and $5000 \text{ BTU/HR}\cdot\text{FT}^2\cdot\text{F}$. An increase in the number of fins results in not only an increase in the area ratio but also an increase in the heat transfer ratio. The increase in the heat transfer ratio is greatest when going from a smooth tube to a tube with 100 fins. The heat transfer ratio increase is greater for the heat transfer coefficient equal to $5000 \text{ BTU/HR}\cdot\text{FT}^2\cdot\text{F}$. This is because the heat transfer coefficient has a direct effect on the heat transfer rate, that is,

$$Q = hA(T - T_{\text{wall}})$$

Both curves approach an asymptotic value. However, the curve with the lower heat transfer coefficient approaches this asymptotic value with a fewer number of fins. Additionally, in view of the relatively small increase in the heat transfer ratio by the addition of fins for the lower heat transfer coefficient, consideration should be given to the cost of manufacturing the fins versus the benefit derived by their addition.



- LEGEND
- = FIN ANGLE = 10
 - = FIN ANGLE = 15
 - △ = FIN ANGLE = 20
 - + = FIN ANGLE = 25

Figure 14. Condensate Level vs. Position (400 Fins, 10-25 Degree Fin Half Angle).

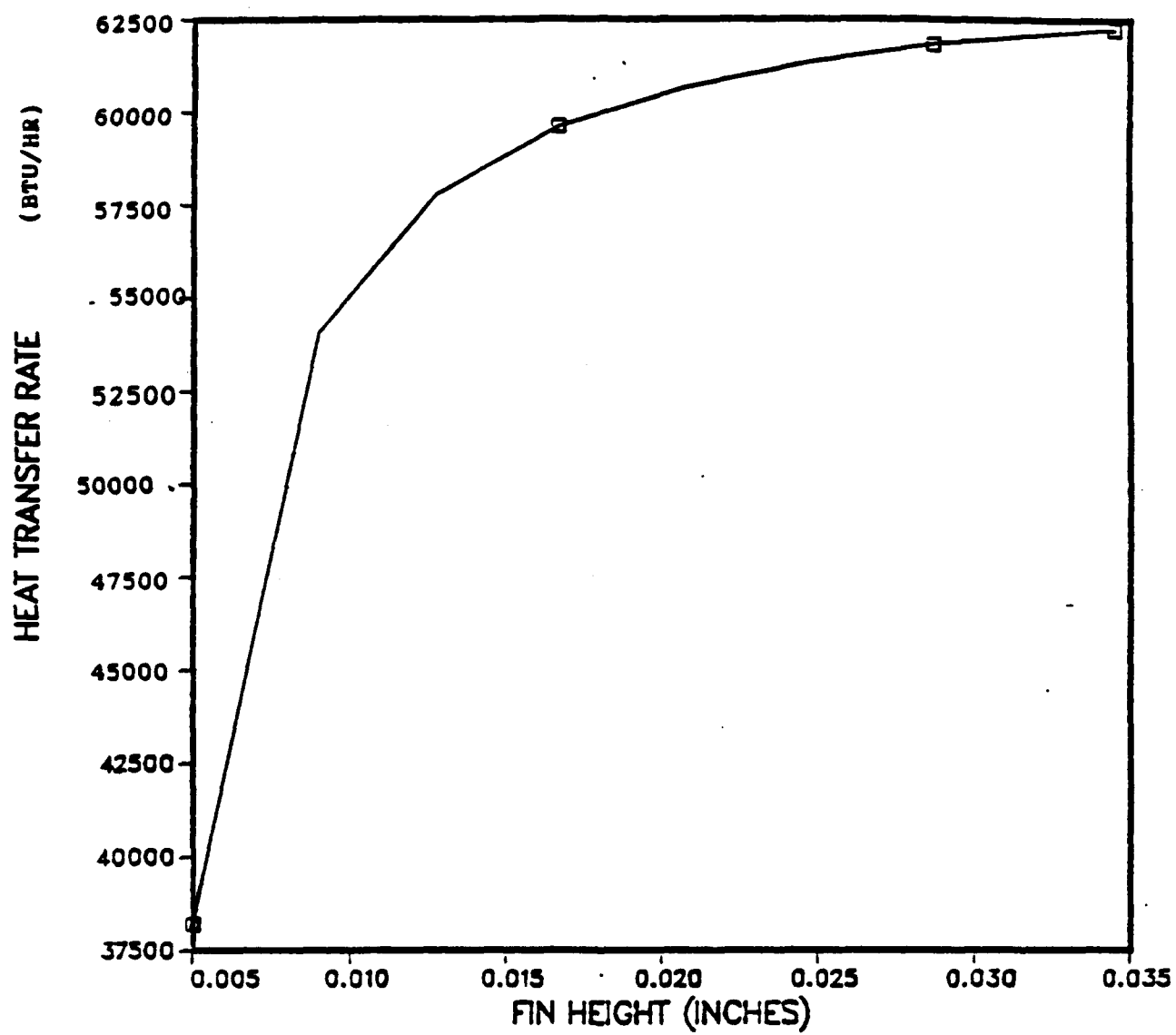


Figure 15. Heat Transfer Rate vs. Fin Height (400 Fins).

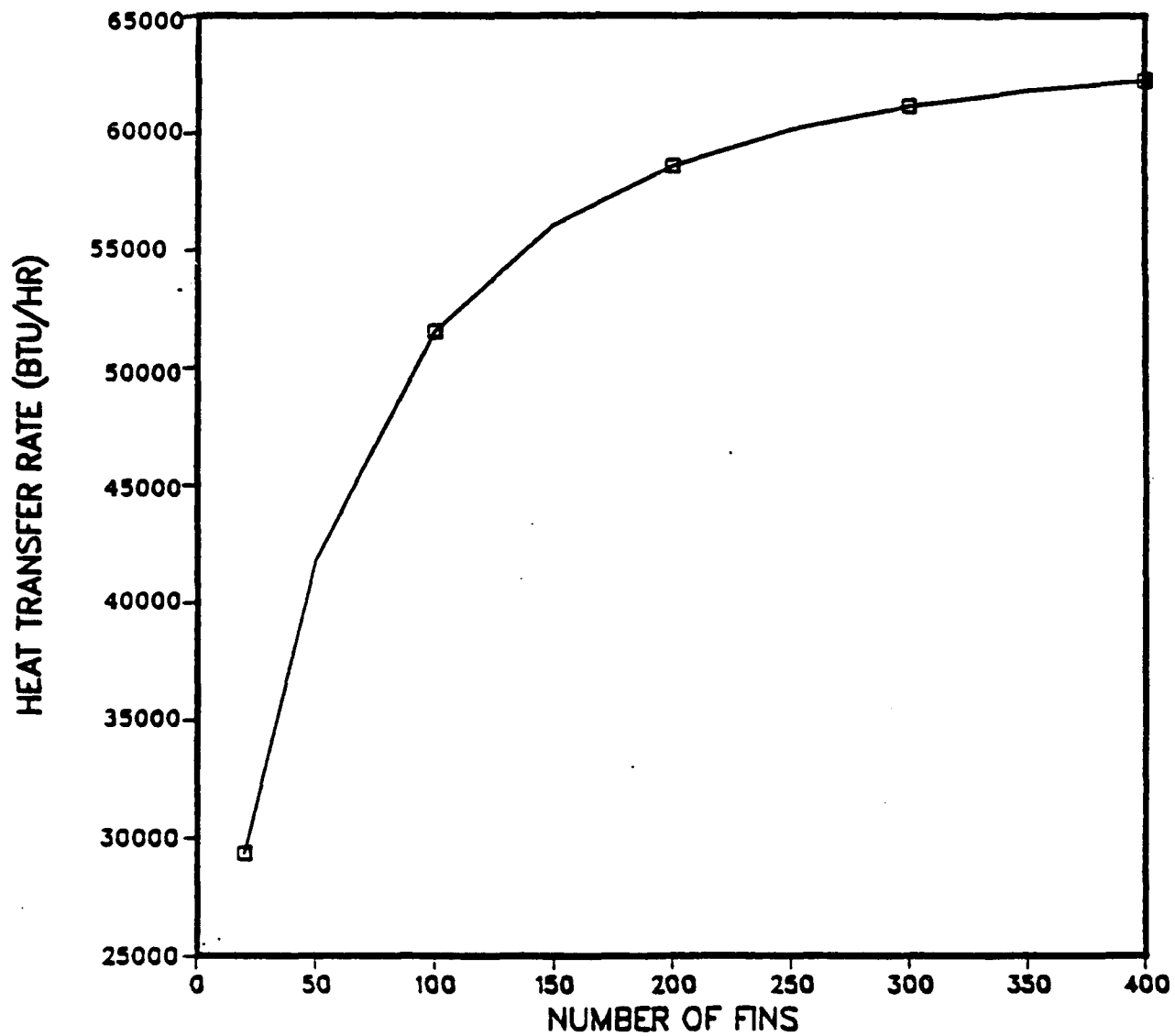
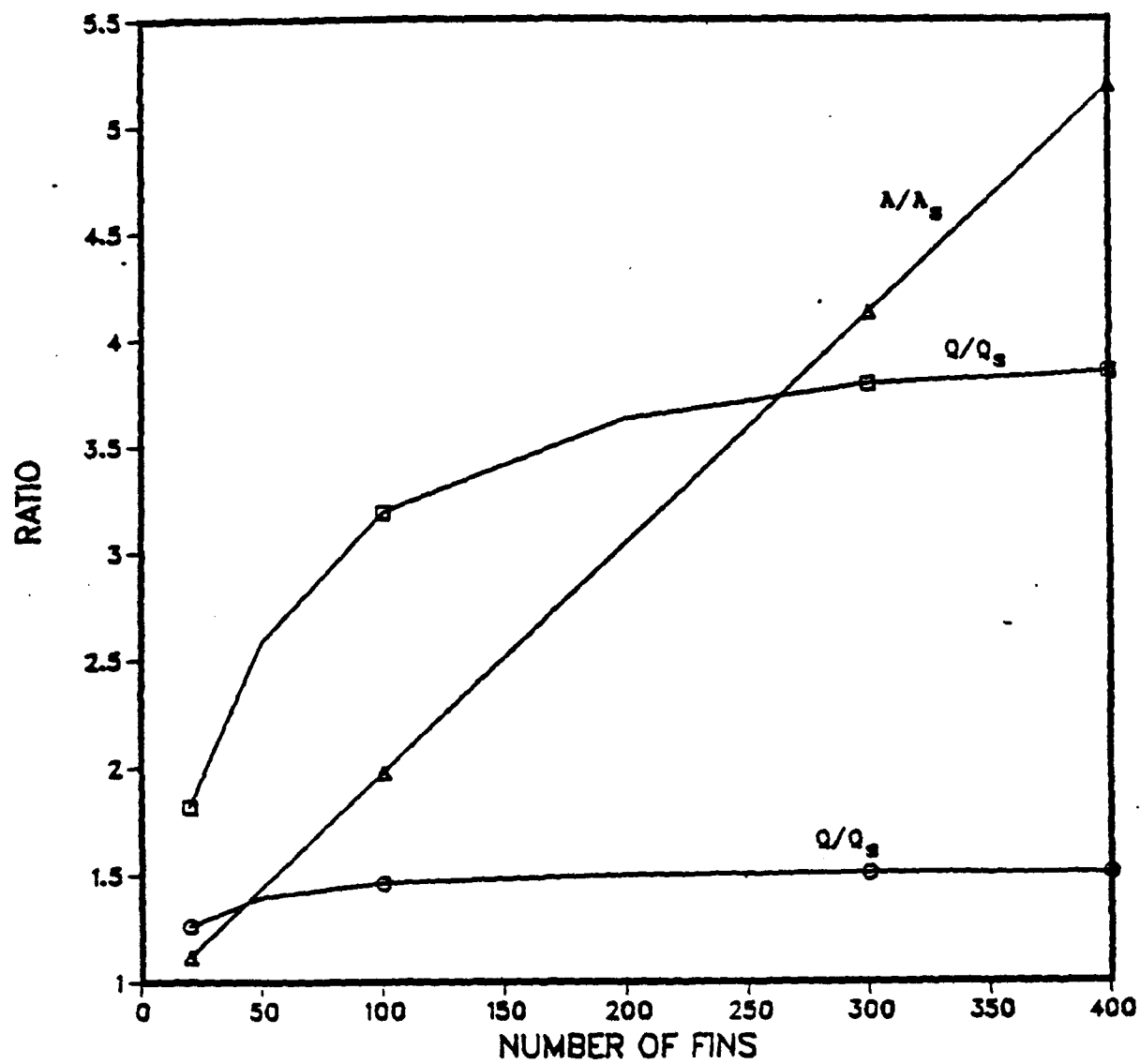


Figure 16. Heat Transfer Rate vs. Number of Fins (Constant Fin Height).



LEGEND
 □ = $H=5000$
 ○ = $H=1000$

Figure 17. Heat Transfer and Area Ratios vs. Number of Fins.

C. AUTOMATED DESIGN SYNTHESIS (ADS)

This optimization project was done on the IBM mainframe using the ADS optimization program. As stated previously, ADS is a general purpose numerical optimization program with a variety of algorithms that can be used to tailor the solution. The solution is separated into three levels in ADS: strategy, optimizer, and one-dimensional search. For this problem the following combination of algorithms were used:

1. Strategy: Sequential Linear Programming
2. Optimizer: Modified Method of Feasible Directions for constrained minimization
3. One-Dimensional Search: Golden Section Method followed by polynomial interpolation

The strategy used linearizes a nonlinear problem by a first order Taylor series expansion of the objective and constraint functions. The solution to this linear approximation is obtained. The problem is linearized again about this point and the new problem is solved with the process being repeated until a precise solution is achieved.

The optimizer chosen is used to find a search direction which will minimize the objective function while maintaining feasibility.

The combination of strategy, optimizer, and one-dimensional search chosen is not the only one available, nor is it necessarily the most efficient. It did yield results that were maximized and were within the constraint tolerances.

The ADS optimization program is complicated by the numerous internal parameters which must be changed to obtain an optimal design.

Complications arise when there is a vast difference in the scales of the design variables. The design variables themselves must be scaled which is further complicated when the variable itself covers a wide range. ADS also, in certain cases, allows for constraints to be violated. In some instances this might be acceptable but not in this case.

ADS does not have a scoping mechanism, that is the ability to decrease the rate of change of the design variable, and therefore to obtain a precise answer the internal parameters must be changed repeatedly. ADS also does not recognize integers, all numbers are real therefore, depending on the answer given, it may be necessary to round up or down.

V. CONCLUSIONS

1. For an independent increase in fin half angle, rotational speed or number of fins an increase is seen in the heat transfer rate. As the parameters increase, the heat transfer rate levels off at the theoretical maximum heat transfer rate for the heat pipe. A decrease in the fin half angle with a corresponding increase in the fin height increases the heat transfer rate. If the fin half angle is decreased while the fin height is kept constant, then the heat transfer rate decreases.

2. Maximum heat transfer occurs for the same fin geometry regardless of the external heat transfer coefficient. For a specific condenser radius, as many fins as possible should be machined with a minimum fin half angle at the maximum fin height.

3. The computer code can be used for single analysis or the automated design of an internally finned rotating heat pipe.

4. The benefit of adding fins is dependent on the external heat transfer coefficient. Consideration of the cost of manufacturing the fins versus the increase in the heat transfer rate should be made.

VI. RECOMMENDATIONS

1. Analyze different shaped fins including rectangular and curved.
2. Modify the code to allow for simultaneous variations of more than one variable.
3. Use different working fluids and heat pipe materials to see if a different internal geometry occurs for the maximum heat transfer rate.
4. Modify the code to use the DOT optimization program vice ADS.

APPENDIX: PROGRAM LISTING

PROGRAM OLSON

```

C *****
C *
C * ANALYSIS OF ROTATING HEAT PIPE , USING TRIANGULAR
C * ELEMENT MODEL
C * COMPILED BY MAJOR IGNATIUS.S.PURNOMO IN JUNE 1978
C *
C * MODIFIED TO PERMIT NUMERICAL OPTIMIZATION
C * USING COPEs/CONMIN
C * BY LCDR WILLIAM A. DAVIS, JR. IN SEPTEMBER 1980
C * MODIFIED TO PERMIT USE OF THE ADS OPTIMIZATION
C * ROUTINE BY LT. G.L. OLSON IN JUNE 1992
C *
C *
C *****

```

CHARACTER*20 NAME

```

*
* COMMON/ADS/DF(21),G(10),IDG(100),IGRAD,INFO,IOPT,IONED,IPRINT,
* :ISTRAT,IWK(2000),IZ(30),OBJ,S(2),VLB(2),VUB(2),W(21,30),WK(5000),
* :NCOLA,NCON,NDV,NGT,NRA,NRIWK,NRWK
*
C
C COMMON/OLLIE/A(200,50),AMTOT(200),APS,B(3),BFIN,BOA,BVIN,C(3),
C :CANGL,CF(200),CK,CLI,COF(5),CW(200),DEL(200),DMDOT(200),EA(3,3),
C :EPS(200),EZERO,F(200,1),FANGL,FNOBJ(100),H(200),HINF,HZ(200),
C :QB(200),QINC(200),QTINC(200),QTOT,QTOTAL(100),R(200),RB(200),
C :RBASEI,R2I,RHOF(200),ROOTI(4),RPM,ROOTR(4),T(200),TALFA,TB(200),
C :TCC(200),TE(200),THICK,THICKI,TIB(200),TINF,TS(200),TSAT,TSS,
C :TT(200),TZ,UF(200),X(200),XCOF(5),XPLOT(200),Y(200),Z(200),ZOA,
C :DOBF,DOTH,ICOR(200,3),IFF,JINT,JLC,JTC,KFF(50),KFIN(50),KT,NBAN,
C :NEL,NFIN,NSNP
*
*

```

GUIDE TO FORTRAN VARIABLE NAMES

```

C AFOVAS FIN AREA/SMOOTH AREA
C ALFA FIN HALF ANGLE (RADIANs)
C BFIN HEIGHT OF FIN (INCHES)
C BOA TANGENT OF THE FIN HALF ANGLE
C BVIN HEIGHT OF FIN (FEET)
C CALFA COSINE OF ALFA
C CANGL CONE HALF ANGLE (DEGREES)
C CBASE INSIDE CIRCUMFERENCE OF CONDENSER (FEET)
C CEXIT INSIDE CIRCUMFERENCE AT CONDENSER EXIT (FEET)
C CF THERMAL CONDUCTIVITY OF CONDENSATE FILM (BTU/HR FT F)
C CL CONDENSER LENGTH (FEET)
C CLI CONDENSER LENGTH (INCHES)
C CPHI COSINE OF PHI
C CRIT CONVERGENCE CRITERION

```

C	DEL	FILM THICKNESS
C	DF	GRADIENT OF OBJECTIVE
C	DIV	FLOATING POINT VALUE OF NDIV
C	DMTOT	CONDENSATE MASS FLOW RATE
C	EPS	TROUGH WIDTH INCLUDING INCREMENTAL CHANGE
C	EPSEX	TROUGH WIDTH AT CONDENSER EXIT
C	EPSO	TROUGH WIDTH AT START OF CONDENSER
C	EZERO	FIN BASE WIDTH
C	F	FORCE VECTOR OF SYSTEM
C	FANGL	FIN HALF ANGLE (DEGREES)
C	G	CONSTRAINT VALUES ASSOCIATED WITH CURRENT DESIGN
C	H	CONVECTIVE HEAT TRANSFER COEFFICIENT (BTU/HR FT ² F)
C	HFG	LATENT HEAT OF VAPORIZATION (BTU/LBM)
C	IDG	CONSTRAINT TYPE IDENTIFIER
C	IEL	THE ELEMENT NUMBER
C	IFF	NO. OF ROWS MINUS ONE OF THE UPPER TRIANGULAR FIN
C	IFIN	EQUALS 0 FOR COPPER, AND 1 FOR STAINLESS STEEL
C	IFLUID	EQUALS 0 FOR WATER, AND 1 FOR FREON
C	IGRAD	GRADIENT CALCULATION IDENTIFIER
C	INFO	CONTROL PARAMETER
C	IONED	ONE DIMENSIONAL SEARCH IDENTIFIER
C	IOPT	OPTIMIZER IDENTIFIER
C	IPRINT	A FOUR DIGIT PRINT CONTROL
C	ISTRAT	OPTIMIZATION STRATEGY IDENTIFIER
C	JINT	NO. OF COLUMNS PLUS ONE BELOW TRIANGULAR FIN
C	JLC	NUMBER OF SYSTEM NODAL POINT LOCATED AT THE CENTER OF SYSTEM COORDINATE
C	JTC	NUMBER OF SYSTEM NODAL POINT LOCATED AT THE JUNCTION OF THE SYMMETRY BOUNDARY AND THE LINE OF INTERSECTION BETWEEN THE FIN AND THE CONDENSER WALL
C	KFF	NUMBER OF SYSTEM NODAL POINTS LOCATED ALONG THE FIN CONVECTIVE BOUNDARY
C	KFIN	NUMBER OF SYSTEM NODAL POINTS LOCATED ON THE SYMMETRIC BOUNDARY OF TRIANGULAR FIN SECTION NOT COUNTING POINTS AT BASE AND APEX
C	KT	NUMBER OF COLUMNS WITHIN THE TROUGH WALL SECTION
C	NBAN	SYSTEM BAND WIDTH
C	NBOTF	LAST ELEMENT AT BOTTOM SIDE
C	NBOTI	FIRST ELEMENT AT BOTTOM SIDE
C	NCON	NUMBER OF CONSTRAINTS
C	NDOBF	NUMBER OF ROWS WITHIN THE FIN
C	NDOTH	NUMBER OF ROWS WITHIN THE TROUGH
C	NDIV	NUMBER OF INCREMENT
C	NDV	NUMBER OF DESIGN VARIABLES
C	NEFB	ELEMENT NUMBER AT BASE OF FIN
C	NEL	NUMBER OF ELEMENTS
C	NEST	ELEMENT NUMBER AT END OF TROUGH
C	NRA	NUMBER OF ROWS IN ARRAY A
C	NRWK	DIMENSIONAL SIZE OF WK
C	NSNP	NUMBER OF SYSTEM NODAL POINTS
C	OBJ	VALUE OF THE OBJECTIVE FUNCTION ASSOCIATED WITH X
C	OMEGA	ROTATIONAL SPEED OF HEAT PIPE (RAD/HR)
C	PHI	CONE HALF ANGLE (RADIAN)
C	PI	PI
C	R2I	DISTANCE FROM CENTERLINE OF THE HEAT PIPE TO HALF THE FIN HEIGHT
C	RBASE	INSIDE RADIUS OF CONDENSER BASE (FEET)
C	RBASEI	INSIDE RADIUS OF CONDENSER BASE (INCHES)
C	REXIT	INSIDE RADIUS OF CONDENSER EXIT (FEET)

```

C   RPM      REVOLUTIONS PER MINUTE
C   S        VECTOR OF DESIGN VARIABLES
C   SALFA    SINE OF ALFA
C   SPHI     SINE OF PHI
C   SURFAR   SURFACE AREA
C   THICK    CONDENSER WALL THICKNESS (FEET)
C   THICKI   CONDENSER WALL THICKNESS (INCHES)
C   TPHI     TANGENT OF PHI
C   TZ       AMBIENT TEMPERATURE
C   UF       VISCOSITY
C   VLB      LOWER BOUNDS ON THE DESIGN VARIABLES
C   VUB      UPPER BOUNDS ON THE DESIGN VARIABLES
C   WK       REAL WORK ARRAY
C   ZFIN     NUMBER OF FINS
C   ZOA      RATIO OF TROUGH WIDTH TO FIN BASE WIDTH
C   ZSTAR    SURFACE LENGTH OF THE FIN MINUS THE SURFACE LENGTH
C           COVERED BY THE CONDENSATE IN THE TROUGH
C   ZZERO    SURFACE LENGTH OF FIN
C

```

```

      PRINT*, 'INPUT FILE NAME'
      READ*, NAME
      OPEN(10,FILE=NAME)
      OPEN(15,FILE='/HTPIPE OUTPUT')
      OPEN(14,FILE='/DUMP OUTPUT')
      OPEN(13,FILE='/GRAPH OUTPUT')

```

```

*
*
C   THE FOLLOWING READS INPUT DATA, PERFORMS HEAT
C   TRANSFER ANALYSIS, AND PRINTS RESULTS.
*
*
*
*
*
*

```

```

C   ***** INPUT MODE *****
*

```

ELEMENT CONNECTIVITIES

```

      READ (10,420) NEL,NSNP,NBAN,IFLUID,IFIN
      WRITE (15,430) NEL,NSNP,NBAN
      WRITE (15,435) IFLUID,IFIN
      WRITE (15,436)
      WRITE (15,437)
      READ (10,440) (ICL,(ICOR(IEL,I),I=1,3),IEL=1,NEL)
      WRITE (15,450)

```

THE CONDENSER GEOMETRY

```

      READ (10,460) CLI,CANGL,RBASEI,R2I,THICKI,BFIN,TZ
      WRITE (15,470) CLI,CANGL,RBASEI,R2I,THICKI,BFIN,TZ
      READ (10,480) NDI,NEST,NEFB,NBOTI,NBOTF
      WRITE (15,490) NDI,NEST,NEFB,NBOTI,NBOTF

```

DATA FOR RUNNING

```

      READ (10,500) RPM,TSS,TINF,HINF

```


WRITE (15,510) RPM,TSS,TINF,HINF

THE CONVERGENCE CRITERIAN

READ (10,520) CRIT
WRITE (15,530) CRIT

INTERNAL FIN GEOMETRY

READ (10,540) FANGL,NFIN
WRITE (15,550) FANGL,NFIN
WRITE(*,*) FANGL,NFIN
READ (10,560) IFF
WRITE (15,570) IFF
READ (10,580) (KFIN(I),KFF(I),I=1,IFF)
READ (10,590) NDOBF,NDOTH,JTC,JLC,JINT,KT
NHB=NEFB/2
NBF=NBOTF+1
DOBF = FLOAT(NDOBF)
DOTH = FLOAT(NDOTH)
WRITE (15,600) ICOR(NBOTI,2),ICOR(NEFB,1),ICOR(NEST,1),
1ICOR(NBOTF,1)

SET CONSTRAINTS

NRA=21
NCOLA=30
NRWK= 5000
NRIWK=2000
NDV= 1
NCON= 2
IGRAD= 0

INITIAL DESIGN

S(1)= BFIN

BOUNDS

VLB(1)= 0.0000001
VUB(1)= 0.75

IDENTIFY CONSTRAINTS

NONLINEAR CONSTRAINT

IDG(1)=1

LINEAR CONSTRAINT

IDG(2)=2

PRINT*, 'INPUT THE VALUES FOR ISTRAT,IOPT,IONED AND IPRINT'

READ*, ISTRAT,IOPT,IONED,IPRINT

INITIALIZE COUNTER

NO=0.0

CHANGE THE INTERNAL PARAMETERS

INFO=-2

CALL DADS(INFO,ISTRAT,IOPT,IONED,IPRINT,IGRAD,NDV,NCON,S,VLB,
:VUB,OBJ,G,IDG,NGT,IZ,DF,W,NRA,NCOLA,WK,NRWK,IWK,NRIWK)

IWK(2)=0

IWK(3)=200

IWK(5)=4

IWK(7)=500

WK(3)=-0.5

```

WK(6)=-0.01
WK(8)=0.05
WK(9)=0.50
WK(10)=0.05
WK(11)=0.005
WK(13)=0.001
WK(14)=0.0001
WK(21)=0.004
WK(22)=0.002
WK(26)=0.004
WK(37)=0.0000000001

```

```

*
10 CALL DADS(INFO,ISTRAT,IOPT,IONED,IPRINT,IGRAD,NDV,NCON,S,VLB,
:VUB,OBJ,G,IDG,NGT,IZ,DF,W,NRA,NCOLA,WK,NRWK,IWK,NRIWK)
IF (INFO.EQ.0) GO TO 360

```

```

*
C ***** EXECUTION MODE *****
*

```

```

NO=NO+1

```

```

*
C CONVERT UNITS OF ALL DIMENSIONAL PARAMETERS
C TO FEET. CONVERT UNITS OF ANGLES TO RADIANS.

```

```

R2I=RBASEI-(S(1)/2)
CL=CLI/12.0
R2=R2I/12.0
RBASE=RBASEI/12.0
BVIN=S(1)/12.0
DIV=FLOAT(NDIV)
PI=3.14159265358979
PHI=2.0*CANGL*PI/360.0
SPHI=SIN(PHI)
CPHI=COS(PHI)
TPHI=TAN(PHI)
DELX=CL/DIV
CBASE=2.0*PI*RBASE
REXIT=RBASE+CL*TPHI
CEXIT=2.0*PI*REXIT
THICK=THICKI/12.0
ALFA=FANGL*2.0*PI/360.0
SALFA=SIN(ALFA)
CALFA=COS(ALFA)
TALFA=TAN(ALFA)
EZERO=2.0*(S(1)/12.0)*TALFA
ZOA=((CBASE-(EZERO*NFIN))/NFIN)/EZERO

```

```

*
C BOUNDARY CONDITIONS AND TEMPERATURE ESTIMATES
C ALONG THE FIN BOUNDARY
C

```

```

DO 20 NTINF=NBOTI,NBOTF
20 TS(NTINF)=TINF
DO 30 NNT=NBFI,NEL
TS(NNT)=0.0
30 H(NNT)=0.0
DO 40 IGT=1,NEST
IE=ICOR(IGT,2)
40 T(IE)=TZ
IG=ICOR(NEST,1)
T(IG)=TZ
*
OMEGA IS IN RADIANS/HOUR
OMEGA=RPM*2.0*PI*60.0

```

```

DO 50 KL=NBOTI,NBOTF
50 H(KL)=HINF
   HIFN=HINF
   TSAT=TSS
   EPSO=ZOA*EZERO
   BOA=TALFA
   SURFAR=NFIN*(2.0*(S(1)/(12*CALFA))+EPSO)
   EPSEX=(CEXIT-(NFIN*EZERO))/NFIN
   BETA=(EPSEX-EPSO)/DIV
   ZZERO=(S(1)/12)/CALFA
   AFOVAS=(ZOA+(1./SALFA))/(1.+ZOA)
   ZA=0.0
   DO 60 NSAT=1,NEST
60 TS(NSAT)=TSAT
   TSOLID=(TSAT+TINF)/2.0
C   TEMPORARY CHANGE - TFILM
   ASMOOTH=0.0
   ACASE=0.0
   QT=0.0
   OBJ=0.0
   QT1=0.0
   QTF=0.0
   QTRF=0.0
   QTOT=0.0
   DMTOT=0.0
   NK=NDIV+1
   DO 350 NI=1,NK
C   R IS THE INCREMENTAL CHANGE IN THE RADIUS OF THE CONDENSER
   R(NI)=R2+NI*DELX*SPHI
   RB(NI)=RBASE+NI*DELX*SPHI
C   EPS IS THE INCREMENTAL CHANGE IN THE TROUGH WIDTH
   EPS(NI)=EPSO+NI*BETA
   APS=EPS(NI)
C
C   NODAL POINT COORDINATES
C
   CALL COORD
65 Z(1)=ZA
   DO 70 IZEL=1,NEFB
   NA=ICOR(IZEL,1)
   NB=ICOR(IZEL,2)
   XE=X(NA)-X(NB)
   YE=Y(NA)-Y(NB)
   ELZ=SQRT(XE**2+YE**2)
70 Z(IZEL+1)=Z(IZEL)+ELZ
   XZB=X(ICOR(NHB,1))-X(ICOR(1,2))
   YZB=Y(ICOR(NHB,1))-Y(ICOR(1,2))
   ZB=SQRT(XZB**2+YZB**2)
   ZC=ZZERO
   IM=1
C
C   PARABOLIC TEMPERATURE DISTRIBUTION ALONG THE FIN
C   BOUNDARY, USING LAGRANGE INTERPOLATION
C
80 TP1=T(ICOR(1,2))
   TP2=T(ICOR(NHB,1))
   TP3=T(ICOR(NEFB,1))
   AP1=TP1/(ZB*ZC)
   AP2=TP2/(ZB*(ZB-ZC))
   AP3=TP3/(ZC*(ZC-ZB))

```

```

BP1=-(ZB+ZC)*AP1
BP2=-ZC*AP2
BP3=-ZB*AP3
A1=AP1+AP2+AP3
B1=BP1+BP2+BP3
TC=0.0
DO 90 NY=1,NEST
90 TC=TC+T(ICOR(NY,2))
AY=FLOAT(NY+1)
TF=(TC+T(ICOR(NY,1))+AY*TS(NY))/(2.0*AY)

```

SOLID-FLUID PROPERTIES

WATER PROPERTIES

```

IF(IFLUID.EQ.1) GO TO 91
HFG=1093.88-0.5703*TS(1)+0.00012819*(TS(1)**2)
1-HFG=0.0000008824*(TS(1)**3)
RHOF(NI)=62.774-0.00255698*TF-0.000053572*TF**2
CF(NI)=0.3034+0.000738927*TF-0.00000147321*TF**2
UF(NI)=0.001397-0.000014669*TF+0.0000000631253*TF**2-0.00000
100000976569*TF**3

```

FREON PROPERTIES

```

91 IF(IFLUID.EQ.0) GO TO 92
HFG=69.5459-0.0156011*TS(1)-0.000455294*(TS(1)**2)+0.00000104144*(
1*TS(1)**3)
RHOF(NI)=102.059-0.025364*TF-0.000502649*(TF**2)+0.00000135407*(TF
1**3)
CF(NI)=0.0594858-0.000429765*TF+0.00000348218*TF**2-0.000000010416
18*TF**3
UF(NI)=0.00078-0.00000525*TF+0.0000000125*TF**2
92 UF(NI)=3600*UF(NI)
IF(IFIN.EQ.1) GO TO 93
CW(NI)=231.7772-0.02222*TSOLID
93 IF(IFIN.EQ.0) GO TO 94
CW(NI)=8.776+0.00265*TSOLID
94 CK=CW(NI)
CONST=RHOF(NI)**2*OMEGA**2*HFG*CPHI*CALFA*R(NI)

```

INITIAL FILM THICKNESS

```

IF (NI.GT.1) GO TO 100
DEL(1)=1.107*(((TSAT-TINF)*CF(NI))/(UF(NI)*HFG))**.25)*((UF(NI)/(
1RHOF(NI)*OMEGA))**.5)
100 CONTINUE

```

AVERAGE ELEMENT CONVECTIVE COEFFICIENT ALONG THE FIN BOUNDARY

```

ZSTAR=ZZERO-DEL(NI)/CALFA
AZZ=DEL(NI)/SALFA
ZZ=ZSTAR
HDEN=(( -A1*ZZ**3)/3.0-(B1*ZZ**2)/2.0)+ZZ*(TSAT-T(1))
AZS=ABS(4*CF(NI)*UF(NI)*HDEN/CONST)**0.25
HAC=0.0
DO 190 IEL=1,NEFB
AZ=Z(IEL)
BZ=Z(IEL+1)
IF (ZSTAR.LE.BZ) GO TO 110

```

```

      GO TO 120
110  IF (HAC.NE.0.0) GO TO 180
      BZ=ZSTAR
120  IF (IEL.NE.1) GO TO 130
      AK=(BZ-AZ)/5.0
      ZZ=AK
      GO TO 140
130  AK=(BZ-AZ)/4.0
      ZZ=AZ
140  ZEL=4*AK
      DO 150 NH=1,5
      HDEN=(-1.0*(A1*ZZ**3/3.0+B1*ZZ**2/2.0))+ZZ*(TSAT-T(1))
      HZ(NH)=ABS(CF(NI)**3*CONST/(4*UF(NI)*HDEN))**0.25
150  ZZ=ZZ+AK
C    AVERAGE H USING SIMPSONS RULE
      CONH=AK*(HZ(1)+4*HZ(2)+2*HZ(3)+4*HZ(4)+HZ(5))/(3*ZEL)
      IF (ZSTAR.EQ.BZ) GO TO 160
      H(IEL)=CONH
      GO TO 190
160  AZ=ZSTAR
      HAZ=CONH*(AZ-Z(IEL))
      DELA=AZS
170  BZ=Z(IEL+1)
      DELB=(BZ-ZSTAR)*AZZ/(ZZERO-ZSTAR)
      DELZ=(DELA+DELB)/2.0
      HAC=(BZ-AZ)*CF(NI)/DELZ
      H(IEL)=(HAZ+HAC)/(BZ-Z(IEL))
      GO TO 190
180  AZ=Z(IEL)
      DELA=DELB
      HAZ=0.0
      GO TO 170
190  CONTINUE
      NETI=NEFB+1
      DO 200 IEL=NETI,NEST
200  H(IEL)=CF(NI)/DEL(NI)
C
C      ENTRY INTO THE FINITE ELEMENT SOLUTION
C
      CALL FORMAF
      CALL BANDEC (NSNP,NBAN,1)
C
C      THE TEMPERATURE DISTRIBUTION
C
      DO 210 NT=1,NSNP
210  T(NT)=F(NT,1)
      TIB(NI)=T(ICOR(NBOTI,2))
      TT(NI)=T(ICOR(NEFB,1))
      TE(NI)=T(ICOR(NEST,1))
      TB(NI)=T(ICOR(NBOTF,1))
      TTS=0.0
      DO 220 NS=1,NSNP
220  TTS=TTS+T(NS)
      PN=FLOAT(NS)
      TSOLID=TTS/PN
C
C      Q AT THE BOTTOM SIDE
C
      QBI=0.0
      DO 230 IBEL=NBOTI,NBOTF

```

```

NKA=ICOR(IBEL,1)
NKB=ICOR(IBEL,2)
XB=X(NKA)-X(NKB)
YB=Y(NKA)-Y(NKB)
ELB=SQRT(XB**2+YB**2)
230 QBI=QBI+(T(NKA)+T(NKB)-2*TS(IBEL))*ELB*H(IBEL)/2.0
QB(NI)=QBI*DELX

```

C
C
C

ITERATION UNTIL CONVERGENCE CRITERIA IS MET

```

IF (IM.EQ.1) GO TO 240
QJ=QBI
GO TO 250
240 QI=QBI
IM=2
GO TO 80
250 AQ=ABS(QJ-QI)/QJ
IF (AQ.LE.CRIT) GO TO 260
QI=QJ
GO TO 80
260 DMDOT(NI)=2.*QBI*DELX/HFG
DMTOT=DMTOT+DMDOT(NI)
C1=RHOF(NI)**2*OMEGA**2*R(NI)*SPHI/(3*UF(NI))
XCOF(1)=-DMTOT
XCOF(2)=0.0
XCOF(3)=0.0
XCOF(4)=C1*EPS(NI)
XCOF(5)=C1*TALFA
M=4
CALL DPOLRT (M,IER)
IF (ROOTR(1).GT.0.0) GO TO 270
IF (ROOTR(2).GT.0.0) GO TO 280
IF (ROOTR(3).GT.0.0) GO TO 290
IF (ROOTR(4).GT.0.0) GO TO 300

```

C
C
C

THE CONDENSATE THICKNESS

```

270 DEL(NI+1)=ROOTR(1)
GO TO 310
280 DEL(NI+1)=ROOTR(2)
GO TO 310
290 DEL(NI+1)=ROOTR(3)
GO TO 310
300 DEL(NI+1)=ROOTR(4)
310 QEL=0.0
IF (NI.NE.1) GO TO 320

```

C
C
C
C
C

Q FROM THE TOP SIDE

Q THROUGH FIN

```

QEL=0.0
320 DO 330 IQEL=1,NEFB
KA=ICOR(IQEL,1)
KB=ICOR(IQEL,2)
XQEL=X(KA)-X(KB)
YQEL=Y(KB)-Y(KA)
ELM=SQRT(XQEL**2+YQEL**2)
QEL=QEL+(2*TS(IQEL)-T(KA)-T(KB))*ELM*H(IQEL)/2.0
330 CONTINUE

```

```

QINC(NI)=QEL*DELX
AMTOT(NI)=DMTOT
QET=QEL*DELX*NFIN*2
QT=QT+QET
QA=QBI*DELX*NFIN*2

C
C
C
Q THROUGH TROUGH

QTRF=0.0
DO 340 IQEL=NEFB+1,NEST
KA=ICOR(IQEL,1)
KB=ICOR(IQEL,2)
XQEL=X(KA)-X(KB)
YQEL=Y(KB)-Y(KA)
ELM=SQRT(XQEL**2+YQEL**2)
QTRF=QTRF+(2*TS(IQEL)-T(KA)-T(KB))*ELM*H(IQEL)/2.0
340 CONTINUE
QTINC(NI)=QTRF*DELX
QTOTAL(NI)=QINC(NI)+QTINC(NI)
QTRFT=QTRF*DELX*NFIN*2.
QTF=QTF+QTRFT
QTOT=QTOT+QA
ASMOOTH=2*PI*RB(NI)*DELX+ASMOOTH
ACASE=ACASE+NFIN*((2*S(1))/(12*CALFA))+EPSO)*DELX
350 CONTINUE
EVALUATE OBJECTIVE FUNCTIONS AND CONSTRAINTS
OBJ=-(QTOT)
WRITE(15,*) 'OBJECTIVE FUNCTION=', OBJ, 'BFIN=', S(1)
C
FIRST CONSTRAINT IS TO ENSURE THE RATIO ZOA IS NOT NEGATIVE
G(1)=-(((CBASE-(2*NFIN*S(1)*TALFA/12))/NFIN)/(S(1)*TALFA/6))
G(1)=1000.0*G(1)
C
THE SECOND CONSTRAINT IS TO ENSURE THE CONDENSATE LEVEL IS NOT
C
GREATER THAN THE FIN HEIGHT
G(2)=-((S(1)/12.0)-DEL(NI))
XPLOT(NO)=S(1)
FNOBJ(NO)=OBJ
ARATIO=ACASE/ASMOOTH
WRITE(13,*) XPLOT(NO),FNOBJ(NO)
WRITE(14,*) ARATIO, FNOBJ(NO)
GO TO 10
360 CONTINUE
C
BFIN=S(1)
C
***** OUTPUT MODE *****
WRITE (15,630)
DO 370 NR=1,NK
370 WRITE (15,640) NR,QINC(NR),QTINC(NR),QTOTAL(NR)
WRITE (15,650) QT,QTF
WRITE (15,660) CLI,CANGL,RBASEI,R2I,THICKI,BFIN,RPM,TSS,TINF,HINF,
1RIT,FANGL,ZOA,IFF
WRITE (15,661) AFOVAS
WRITE (15,670) BOA,ZOA,NFIN,BVIN,SURFAR
WRITE (15,680)
DO 380 NP=1,NSNP
TCC(NP)=.5555555*(T(NP)-32)
380 WRITE (15,690) NP,X(NP),Y(NP),T(NP),TCC(NP)
WRITE (15,700)
DO 390 KKL=1,NBOTF
NKX=ICOR(KKL,1)
NKY=ICOR(KKL,2)

```

```

XP=X(NKX)-X(NKY)
YP=Y(NKX)-Y(NKY)
EXY=SQRT(XP**2+YP**2)
QEP=ABS((T(NKX)+T(NKY)-2*TS(KKL))*EXY*H(KKL)/2.0)
QEP=QEP*DELX
390 WRITE (15,710) KKL,H(KKL),EXY,QEP
WRITE (15,720) CRIT
WRITE (15,730) HFG,NFIN,H(NBOTF),TSAT,RPM,QTOT.QT,FANGL
WRITE(15,734)
WRITE(15,735) ROOTR(1),ROOTI(1)
WRITE(15,735) ROOTR(2),ROOTI(2)
WRITE(15,735) ROOTR(3),ROOTI(3)
WRITE(15,735) ROOTR(4),ROOTI(4)
WRITE (15,740)
DO 400 NR=1,NK
400 WRITE (15,750) NR,DEL(NR),QB(NR),AMTOT(NR),TIB(NR),TT(NR),TE(NR),
1TB(NR)
WRITE (15,760)
DO 410 NG=1,NDIV,2
410 WRITE (15,770) NG,CW(NG),CF(NG),RHOF(NG),UF(NG),EPS(NG),R(NG),
1QINC(NG)
RETURN
C
C 412 FORMAT (8X,E12.5,8X,E12.5)
420 FORMAT (5I5)
430 FORMAT (/2X,15HNO.OF.ELEMENTS=,I5,10X,18HNO.OF.SYSTEM N.P.=,
1I5,/,2X,13HNO.OF.BANDED=,I5)
435 FORMAT (/2X,'IFLUID=',I5,10X,'IFIN=',I5)
436 FORMAT (2X,'IFLUID = 0 FOR WATER, AND 1 FOR FREON')
437 FORMAT (2X,'IFIN = 0 FOR COPPER, AND 1 FOR STAINLESS STEEL')
440 FORMAT (4I5)
450 FORMAT (/2X,7HELEMENT,10X,3HNP1,14X,3HNP2,15X,3HNP3)
460 FORMAT (7G10.5)
470 FORMAT (4X,5HCLI= ,E12.5,/,4X,7HCANGL= ,E12.5,/,4X,8HRBASEI=,E12.
15,/,4X,5HR2I= ,E12.5,/,4X,8HTHICKI= ,E12.5,/,4X,6HBFIN= ,E12.5,/,4
2X,4HTZ= ,E12.5)
480 FORMAT (5I5)
490 FORMAT (4X,6HNDIV= ,I10,/,4X,6HNEST= ,I10,/,4X,6HNEFB= ,I10,/,4X,7
1HNBOTI= ,I10,/,4X,7HNBOTF= ,I10)
500 FORMAT (4F10.2)
510 FORMAT (4X,5HRPM= ,E12.5,/,4X,5HTSS= ,E12.5,/,4X,6HTINF= ,E12.5,/,
14X,6HHINF= ,E12.5)
520 FORMAT (G10.9)
530 FORMAT (4X,6HCRIT= ,E12.5)
540 FORMAT (2G10.5)
550 FORMAT (4X,7HFANGL= ,E12.5,/,4X,6HNFIN= ,I5)
560 FORMAT (I5)
570 FORMAT (4X,5HIFF= ,I10)
580 FORMAT (16I5)
590 FORMAT (6I5)
600 FORMAT (///5X,4HTIB=,I5,10X,3HTT=,I5,/,5X,3HTE=,I5,/,
1,6X,3HTB=,I5)
610 FORMAT (//10X,17HCRASH,CRASH,CRASH)
620 FORMAT (//5X,4(E12.7,3X))
630 FORMAT (2X,7HSTATION,2X,4HQFIN,17X,7HQTROUGH,15X,6HQTOTAL)
640 FORMAT (4X,I5,E12.5,10X,E12.5,10X,E12.5)
650 FORMAT (///,4X,11HQFIN TOTAL=,E12.5,10X,15HQTROUGH TOTAL= ,E12.5)
660 FORMAT (//////,4X,5HCLI= ,E12.5,5X,7HCANGL= ,E12.5,/,4X,8HRBASEI= ,
1E12.5,2X,5HR2I= ,E12.5,/,4X,8HTHICKI= ,E12.5,2X,6HBFIN= ,E12.5,/,4
2X,5HRPM= ,E12.5,5X,5HTSS= ,E12.5,/,4X,6HTINF= ,E12.5,4X,6HHINF= ,E

```



```

312.5,/,4X,6HCRT= ,E12.5,4X,7HFANGL= ,E12.5,/,4X,5HZOA= ,E12.5,5X,
45HIFF= ,I10)
661 FORMAT(4X,'FIN AREA/SMOOTH AREA=',E12.5)
670 FORMAT (1H1,/,2X,4HBOA=,G12.5,5X,4HZOA=,G12.5,3X,5HNFN=,I5,5X,
1/,5HBVIN=,G12.5,5X,13HSURFACE AREA=,G12.5)
680 FORMAT (/,5X,2HNPF,6X,1HX,12X,1HY,12X,1HT,12X,2HTC)
690 FORMAT (/2X,I3,3X,4(F10.6,3X))
700 FORMAT (/,2X,2HEL,8X,1HH,11X,9HEL-LENGTH,15X,4HQ-EL)
710 FORMAT (/2X,I2,3X,E12.5,3X,E12.5,10X,E12.5)
720 FORMAT (/2X,2HCONVERGENCE CRITERIAN=,E15.8)
730 FORMAT (1H ,/,5X,4HHFG=,E12.5,/,5X,11HNO.OF FINS=,I5,/,5X,
16HH-OUT=,E12.5,/,5X,5HTSAT=,E12.5,/,5X,4HRPM=,E12.5,/,5X,6HQ-BOT=,
2E12.5,/,5X,6HQFIN =,E12.5,/,5X,11HHALF-ANGLE=,F8.3)
734 FORMAT(/5X,'ROOTS:',5X,'REAL PARTS',15X,'IMAGINARY PARTS')
735 FORMAT(15X,E12.5,15X,E12.5)
740 FORMAT (1H0,6X,1HJ,4X,14HFILM THICKNESS,8X,2HQB,10X,8HMASS-TOT,
1/,4X,3HTIB,8X,2HTT,10X,2HTE,8X,2HTB)
750 FORMAT (1H ,4X,I4,4X,F12.10,4X,F10.4,6X,F9.5,6X,F5.1,/,
16X,F5.1,6X,F5.1,6X,F5.1)
760 FORMAT (1H0,6X,1HJ,6X,6HK-WALL,4X,6HK-FILM,3X,7HDENSITY,4X,9HVIS-
1FILM,/,6X,7HEPSILON,5X,6HRADIUS,5X,4HQINC)
770 FORMAT (1H ,4X,I4,4X,F7.3,4X,F6.4,4X,F6.3,4X,F9.7,
1/,4X,F9.7,4X,F7.5,5X,F5.1,1X,F7.3)
END

```

THIS SUBROUTINE DEFINES THE POSITIONS OF SYSTEM COORDINATE POINTS

SUBROUTINE COORD

```

COMMON/ADS/DF(21),G(10),IDG(100),IGRAD,INFO,IOPT,IONED,IPRINT,
:ISTRAT,IWK(2000),IZ(30),OBJ,S(2),VLB(2),VUB(2),W(21,30),WK(5000),
:NCOLA,NCON,NDV,NGT,NRA,NRIWK,NRWK

```

```

COMMON/OLLIE/A(200,50),AMTOT(200),APS,B(3),BFIN,BOA,BVIN,C(3),
:CANGL,CF(200),CK,CLI,COF(5),CW(200),DEL(200),DMDOT(200),EA(3,3),
:EPS(200),EZERO,F(200,1),FANGL,FNOBJ(100),H(200),HINF,HZ(200),
:QB(200),QINC(200),QTINC(200),QTOT,QTOTAL(100),R(200),RB(200),
:RBASEI,R2I,RHOF(200),ROOTI(4),RPM,ROOTR(4),T(200),TALFA,TB(200),
:TCC(200),TE(200),THICK,THICKI,TIB(200),TINF,TS(200),TSAT,TSS,
:TT(200),TZ,UF(200),X(200),XCOF(5),XPLOT(200),Y(200),Z(200),ZOA,
:DOBF,DOTH,ICOR(200,3),IFF,JINT,JLC,JTC,KFF(50),KFIN(50),KT,NBAN,
:NEL,NFIN,NSNP

```

DELH IS THE STANDARD DIVISION OF FIN HEIGHT

```

DELH=S(1)/(12*DOBF)
X(1)=0.0
Y(1)=THICK+(S(1)/12)
N=1
DO 20 I=1,IFF
ICA=KFIN(I)
ICB=KFF(I)
CBA=FLOAT(ICB-ICA)
AN=0.0
DO 10 II=ICA,ICB
X(II)=X(1)+N*AN*DELH*TALFA/CBA
Y(II)=Y(1)-N*DELH
10 AN=AN+1.0
20 N=N+1

```

```

AN=0.0
ICD=ICB-ICA+1
DO 50 J=JTC,JLC,JINT
X(J)=X(1)
Y(J)=(1.0-AN/DOTH)*THICK
DO 30 JJ=1,ICD
X(J+JJ)=X(J)+JJ*EZERO/(2*(CBA+1.0))
30 Y(J+JJ)=Y(J)
JJ=ICD
DO 40 K=1,KT
X(J+JJ+K)=X(J+JJ)+K*APS/(2.0*KT)
40 Y(J+JJ+K)=Y(J)
50 AN=AN+1.0
RETURN
END

```

THIS SUBROUTINE IS USED TO FORMULATE THE FEM EQUATIONS

SUBROUTINE FORMAF

```

COMMON/ADS/DF(21),G(10),IDG(100),IGRAD,INFO,IOPT,IONED,IPRINT,
:ISTRAT,IWK(2000),IZ(30),OBJ,S(2),VLB(2),VUB(2),W(21,30),WK(5000),
:NCOLA,NCON,NDV,NGT,NRA,NRIWK,NRWK

```

```

COMMON/OLLIE/A(200,50),AMTOT(200),APS,B(3),BFIN,BOA,BVIN,C(3),
:CANGL,CF(200),CK,CLI,COF(5),CW(200),DEL(200),DMDOT(200),EA(3,3),
:EPS(200),EZERO,F(200,1),FANGL,FNOBJ(100),H(200),HINF,HZ(200),
:QB(200),QINC(200),QTINC(200),QTOT,QTOTAL(100),R(200),RB(200),
:RBASEI,R2I,RHOF(200),ROOTI(4),RPM,ROOTR(4),T(200),TALFA,TB(200),
:TCC(200),TE(200),THICK,THICKI,TIB(200),TINF,TS(200),TSAT,TSS,
:TT(200),TZ,UF(200),X(200),XCOF(5),XPLOT(200),Y(200),Z(200),ZOA,
:DOBF,DOTH,ICOR(200,3),IFF,JINT,JLC,JTC,KFF(50),KFIN(50),KT,NBAN,
:NEL,NFIN,NSNP

```

```

DO 20 N=1,NSNP
F(N,1)=0.0
DO 10 MA=1,NBAN
10 A(N,MA)=0.0
20 CONTINUE
DO 70 IEL=1,NEL
IA=ICOR(IEI,1)
IB=ICOR(IEI,2)
IC=ICOR(IEI,3)
B(1)=Y(IB)-Y(IC)
B(2)=Y(IC)-Y(IA)
B(3)=Y(IA)-Y(IB)
C(1)=X(IC)-X(IB)
C(2)=X(IA)-X(IC)
C(3)=X(IB)-X(IA)
C LENGTH BETWEEN ELEMENT NODES 1 AND 2
EL=SQRT(C(3)**2+B(3)**2)
C AREA OF TRIANGULAR ELEMENT
AS=ABS((B(1)*C(2)-B(2)*C(1))/2.0)
HC=H(IEI)/CK
DO 60 J=1,3
JJ=ICOR(IEI,J)
DO 50 K=1,3
KK=ICOR(IEI,K)

```

```

C      FORMING THE A MATRIX
      EA(J,K)=(B(J)*B(K)+C(J)*C(K))/(4*AS)
      IF (HC.EQ.0.0) GO TO 40
      HEL=HC*EL/6.0
      IF (J.EQ.3) GO TO 40
      IF (K.EQ.3) GO TO 40
      IF (J.EQ.K) GO TO 30
      EA(J,K)=EA(J,K)+HEL
      GO TO 40
30    EA(J,K)=EA(J,K)+2*HEL
40    IF (KK.LT.JJ) GO TO 50
      NW=KK-JJ+1
      A(JJ,NW)=A(JJ,NW)+EA(J,K)
50    CONTINUE
60    CONTINUE
C      FORMING THE F MATRIX
      FE=HC*TS(IEL)*EL/2.0
      F(IA,1)=F(IA,1)+FE
      F(IB,1)=F(IB,1)+FE
70    CONTINUE
      RETURN
      END

*
*
*      EQUATION SOLVER OF A SYMMETRIC MATRIX THAT HAS BEEN TRANS-
*      FORMED INTO BANDED FORM.
*
*      SUBROUTINE BANDEC (NEQ,MAXB,NVEC)
*
*      COMMON/ADS/DF(21),G(10),IDG(100),IGRAD,INFO,IOPT,IONED,IPRINT,
*:ISTRAT,IWK(2000),IZ(30),OBJ,S(2),VLB(2),VUB(2),W(21,30),WK(5000),
*:NCOLA,NCON,NDV,NGT,NRA,NRIWK,NRWK
*
*      COMMON/OLLIE/A(200,50),AMTOT(200),APS,B(3),BFIN,BOA,BVIN,C(3),
*:CANGL,CF(200),CK,CLI,COF(5),CW(200),DEL(200),DMDOT(200),EA(3,3),
*:EPS(200),EZERO,F(200,1),FANGL,FNOBJ(100),H(200),HINF,HZ(200),
*:QB(200),QINC(200),QTINC(200),QTOT,QTOTAL(100),R(200),RB(200),
*:RBASEI,R2I,RHOF(200),ROOTI(4),RPM,ROOTR(4),T(200),TALFA,TB(200),
*:TCC(200),TE(200),THICK,THICKI,TIB(200),TINF,TS(200),TSAT,TSS,
*:TT(200),TZ,UF(200),X(200),XCOF(5),XPLOT(200),Y(200),Z(200),ZOA,
*:DOBF,DOTH,ICOR(200,3),IFF,JINT,JLC,JTC,KFF(50),KFIN(50),KT,NBAN,
*:NEL,NFIN,NSNP
*
*
      LOOP=NEQ-1
      DO 20 I=1,LOOP
      MB=I+1
      NB=MIN0(I+MAXB-1,NEQ)
      DO 20 J=MB,NB
      L=J+2-MB
      D=A(I,L)/A(I,1)
      DO 10 MM=1,NVEC
10    F(J,MM)=F(J,MM)-D*F(I,MM)
      MM=MIN0(MAXB-L+1,NEQ-J+1)
      DO 20 K=1,MM
      NN=L+K-1
20    A(J,K)=A(J,K)-D*A(I,NN)
      DO 30 I=1,NVEC
30    F(NEQ,I)=F(NEQ,I)/A(NEQ,1)
      DO 50 I=2,NEQ

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J=NEQ-I+1
K=MIN0(NEQ-J+1,MAXB)
DO 50 MM=1,NVEC
DO 40 L=2,K
MB=J+L-1
40 F(J,MM)=F(J,MM)-A(J,L)*F(MB,MM)
50 F(J,MM)=F(J,MM)/A(J,1)
RETURN
END

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C SUBROUTINE DPOLRT COMPUTES THE ROOTS OF A REAL
C POLYNOMIAL USING A NEWTON-RAPHSON ITERATIVE
C TECHNIQUE.
C

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SUBROUTINE DPOLRT (M,IER)

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COMMON/ADS/DF(21),G(10),IDG(100),IGRAD,INFO,IOPT,IONED,IPRINT,
:ISTRAT,IWK(2000),IZ(30),OBJ,S(2),VLB(2),VUB(2),W(21,30),WK(5000),
:NCOLA,NCON,NDV,NGT,NRA,NRIWK,NRWK

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```

COMMON/OLLIE/A(200,50),AMTOT(200),APS,B(3),BFIN,BOA,BVIN,C(3),
:CANGL,CF(200),CK,CLI,COF(5),CW(200),DEL(200),DMDOT(200),EA(3,3),
:EPS(200),EZERO,F(200,1),FANGL,FNOBJ(100),H(200),HINF,HZ(200),
:QB(200),QINC(200),QTINC(200),QTOT,QTOTAL(100),R(200),RB(200),
:RBASEI,R2I,RHOF(200),ROOTI(4),RPM,ROOTR(4),T(200),TALFA,TB(200),
:TCC(200),TE(200),THICK,THICKI,TIB(200),TINF,TS(200),TSAT,TSS,
:TT(200),TZ,UF(200),X(200),XCOF(5),XPLOT(200),Y(200),Z(200),ZOA,
:DOBF,DOTH,ICOR(200,3),IFF,JINT,JLC,JTC,KFF(50),KFIN(50),KT,NBAN,
:NEL,NFIN,NSNP

```

```

IFIT=0
N=M
IER=0
IF (XCOF(N+1)) 10,40,10
10 IF (N) 20,20,60

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```

SET ERROR CODE TO 1

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20 IER=1
30 RETURN

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```

SET ERROR CODE TO 4

```

```

40 IER=4
GO TO 30

```

```

SET ERROR CODE TO 2

```

```

50 IER=2
GO TO 30
60 IF (N-36) 70,70,50
70 NX=N
NXX=N+1
N2=1
KJ1=N+1
DO 80 L=1,KJ1
MT=KJ1-L+1

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```

      80 COF(MT)=XCOF(L)
C
C      SET INITIAL VALUES
C
      90 XO=.00500101
        YO=0.01000101
C
C      ZERO INITIAL VALUE COUNTER
C
        IN=0
      100 XX=XO
C
C      INCREMENT INITIAL VALUES AND COUNTER
C
        XO=-10.0*YO
        YO=-10.0*XX
C
C      SET X AND Y TO CURRENT VALUE
C
        XX=XO
        YY=YO
        IN=IN+1
        GO TO 120
      110 IFIT=1
        XPR=XX
        YPR=YY
C
C      EVALUATE POLYNOMIAL AND DERIVATIVES
C
      120 ICT=0
      130 UX=0.0
        UY=0.0
        V=0.0
        YT=0.0
        XT=1.0
        U=COF(N+1)
        IF (U) 140,270,140
      140 DO 150 I=1,N
        L=N-I+1
        XT2=XX*XT-YY*YT
        YT2=XX*YT+YY*XT
        U=U+COF(L)*XT2
        V=V+COF(L)*YT2
        FI=I
        UX=UX+FI*XT*COF(L)
        UY=UY-FI*YT*COF(L)
        XT=XT2
      150 YT=YT2
        SUMSQ=UX*UX+UY*UY
        IF (SUMSQ) 160,230,160
      160 DX=(V*UY-U*UX)/SUMSQ
        XX=XX+DX
        DY=-(U*UY+V*UX)/SUMSQ
        YY=YY+DY
        IF (ABS(DY)+ABS(DX)-1.0E-05) 210,170,170
C
C      STEP ITERATION COUNTER
C
      170 ICT=ICT+1
        IF (ICT-500) 130,180,180

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```

180 IF (IFIT) 210,190,210
190 IF (IN-5) 100,200,200
C
C      SET ERROR CODE TO 3
C
200 IER=3
    GO TO 30
210 DO 220 L=1,NXX
    MT=KJ1-L+1
    TEMP=XCOF(MT)
    XCOF(MT)=COF(L)
220 COF(L)=TEMP
    ITEMP=N
    N=NX
    NX=ITEMP
    IF (IFIT) 250,110,250
230 IF (IFIT) 240,100,240
240 XX=XPR
    YY=YPR
250 IFIT=0
    IF (ABS(YY/XX)-1.0E-04) 280,260,260
260 ALPHA=XX+XX
    SUMSQ=XX*XX+YY*YY
    N=N-2
    GO TO 290
270 XX=0.0
    NX=NX-1
    NXX=NXX-1
280 YY=0.0
    SUMSQ=0.0
    ALPHA=XX
    N=N-1
290 COF(2)=COF(2)+ALPHA*COF(1)
    DO 300 L=2,N
300 COF(L+1)=COF(L+1)+ALPHA*COF(L)-SUMSQ*COF(L-1)
310 ROOTI(N2)=YY
    ROOTR(N2)=XX
    N2=N2+1
    IF (SUMSQ) 320,330,320
320 YY=-YY
    SUMSQ=0.0
    GO TO 310
330 IF (N) 30,30,90
    END

```

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